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DEFINITION OF REVERSE OSMOSIS PUMP REQUIREMENTS FOR SPACE VEHICLE REQUIREMENTS

M. S. Bonura, et al

McDonnell Douglas Astronautics Company

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DEFINITION OF REVERSE OSMOSIS PUMP REQUIREMENTS FOR SPACE VEHICLE APPLICATION

Final Report

JUNE 1973 MDC G4744

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PREPARED FOR THE OFFICE OF SALINE WATER, U.S. DEPARTMENT OF THE INTERIOR, UNDER CONTRACT OSW 14-30-3062

Research and Development Progress Report # 892

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY-WEST
5301 Bolsa Avenue, Huntington Beach, CA 92647

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Definition of Reverse Osmosis Pump Requirements for Space Vehicle Requirements

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McDonnell Douglas Astronautics Co.
West Huntington Beach, California

174 pages, 45 figures, 13 tables

The ~~This~~ report covers the investigation of high-pressure reverse osmosis (RO) pumps for space vehicle application. The vendor pump survey and pump evaluation tasks are described in this report. Four selected RO pumps were ordered and evaluated during a 30-day test under spacecraft wash water recovery system conditions. Of the four candidate pumps, only two successfully completed the required 30 days (720 hours) of operation. A description of the test stand and the data from the pump evaluation test are contained in this report. The test data include pump power requirements, efficiency and other design parameters. The test data were used to develop recommendations for a flight-type prototype RO pump and a design specification for a high-pressure RO pump for space vehicle application. Also included are rough order of magnitude (ROM) costs for the development of a flight type RO pump.

Reverse Osmosis*, Pumps*, Pump Testing*, Centrifugal Pumps*, Pump Performance.

Reciprocating Pumps*, Gear Pumps*, Vane Pumps*, Pump Requirements for Space Vehicle, Reverse Osmosis Pumps, Flight Pump Specification, Pump Design Features Reverse Osmosis Pump Tests

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Lee M. Kindley

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FOREWORD

The work described in this report was performed under the Office of Saline Water, United States Department of Interior, Contract No. OSW 14-30-3062, which was jointly funded by the National Aeronautics and Space Administration. The design and preparation of the test facility was accomplished under the McDonnell Douglas Astronautics Company (MDAC) Independent Research and Development Program.

The authors wish to acknowledge the major contributions that were made to this report by these MDAC personnel:

E. P. Honorof--Chemical Analysis

R. E. Shook—Post Test Pump Evaluation

R. L. Vaughan—Test Stand Automatic Control Design

D. C. Ward—Test Stand Mechanical Design

W. Wong—Test Monitoring and Data Reduction

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Section 1
SUMMARY

This final report covers the investigation of high-pressure reverse osmosis (RO) pumps for space vehicle application done under Contract OSW-14-30-3062. The initial phase of this contract defined RO system requirements for spacecraft wash water recycling and is reported in McDonnell Douglas Astronautics Company (MDAC) Report MDC-G3780, dated November 1972. The vendor pump survey and pump evaluation tasks in the second contract phase are described in this report. Four selected RO pumps were ordered and evaluated during a 30-day test under spacecraft wash water recovery system conditions. Of the four candidate pumps, only two successfully completed the required 30 days (720 hours) of operation. A description of the test stand and the data from the pump evaluation test are contained in this report. The test data include pump power requirements, efficiency and other design parameters. The test data were used to develop recommendations for a flight-type prototype RO pump and a design specification for a high-pressure RO pump for space vehicle application. Also included are rough order of magnitude (ROM) costs for the development of a flight type RO pump.

Section 2
INTRODUCTION

Two pacing components for a reverse osmosis (RO) system for spacecraft water recovery are (1) a small-size, long-life, heat-sterilizable membrane assembly and (2) a high-pressure pump that is lightweight, has low power consumption, and will run for long periods with minimum maintenance. This contract, OSW 14-30-3062, addresses both of these problems in a two-phase program. Work done in the initial phase of the contract defined RO system requirements for spacecraft wash water recycling and was completed in November 1972 with the distribution of report MDC G3780, "Definition of Reverse Osmosis Requirements for Spacecraft Wash Water Recycling." Such items as module design, flow rates, recovery fractions, wash water composition, control methods, tradeoff factors, and design requirements are defined. In the second phase of the contract information on pressures, temperatures, flows, and wash water solutes from the initial phase were used to develop a preliminary design and specification for a flight configuration RO pump. Commercial pumps were evaluated with simulated spacecraft wash water to determine materials compatibility, seal life, design methods, power requirements and other factors to insure that the contents of the design specification cover all aspects of pump requirements for the intended service.

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Section 3
REVERSE OSMOSIS PUMP EVALUATION AND SELECTION

The design of a lightweight, high-pressure pump having high reliability and low power consumption is a pacing item for an RO system for spacecraft water recovery. The objective of the RO pump study was to develop a preliminary design and specification for a flight-configuration, high-pressure pump for a modularized six-man RO wash water recovery system. The pump evaluation and selection task is described in this section and included the following subtasks: preliminary pump requirements were identified; a study was made of available pump types and their capabilities; a survey was made of vendors of commercial pumps to find off-the-shelf pumps satisfying RO pump requirements; a selection was made based on criteria generated to quantitatively evaluate the candidate commercial pumps and four pumps were ordered for testing. The RO pump test results are described in Section 4 of this report. Additional information is presented on these subtasks in the following paragraphs.

3.1 TYPES OF PUMPS

Pumps considered for use in RO systems may be divided into four general categories based on the principle of operation of the pumping mechanism. These categories are (1) reciprocating pumps, (2) centrifugal pumps, (3) vane pumps, and (4) gear pumps. General descriptions, advantages, and operational limitations of these categories of pumps are presented in Subsections 3.1.1 through 3.1.4. The information in these subsections was compiled from References 1 through 7.

3.1.1 Reciprocating Pumps

Reciprocating pumps may be either piston or packed plunger type and may be actuated directly by a power cylinder connected to the piston rod or driven by a crankshaft connected to an external power source. All reciprocating pumps are commonly referred to as piston pumps. Piston pumps have high efficiencies, high reliability, and high power density—maximum power for minimum size and weight. Aircraft and missile systems use piston pumps almost exclusively because of these considerations.

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Piston pumps are in general slightly more expensive and require better filtration of the inlet fluid than other pump types. When correct inlet and outlet conditions are maintained, piston pumps may be expected to give longer trouble-free service than other types of hydraulic pumps.

3.1.1.1 Crankshaft-Driven Piston Pumps

Reciprocating pumps come in a variety of crankshaft-driven designs. A wide range of mechanisms are employed to impart reciprocating motion to the connecting rod and piston assemblies. These pumps are available in both variable and fixed displacement types. A typical crankshaft driven piston pump is shown in Figure 3-1.

Some designs employ a crankshaft-driven piston to displace oil which in turn moves a diaphragm in contact with the working fluid. This type of design is widely used in the chemical and process industries and precise control of output volume may be achieved. Since only the cylinder head and diaphragm are in contact with the fluid to be pumped, the problems of lubrication of the

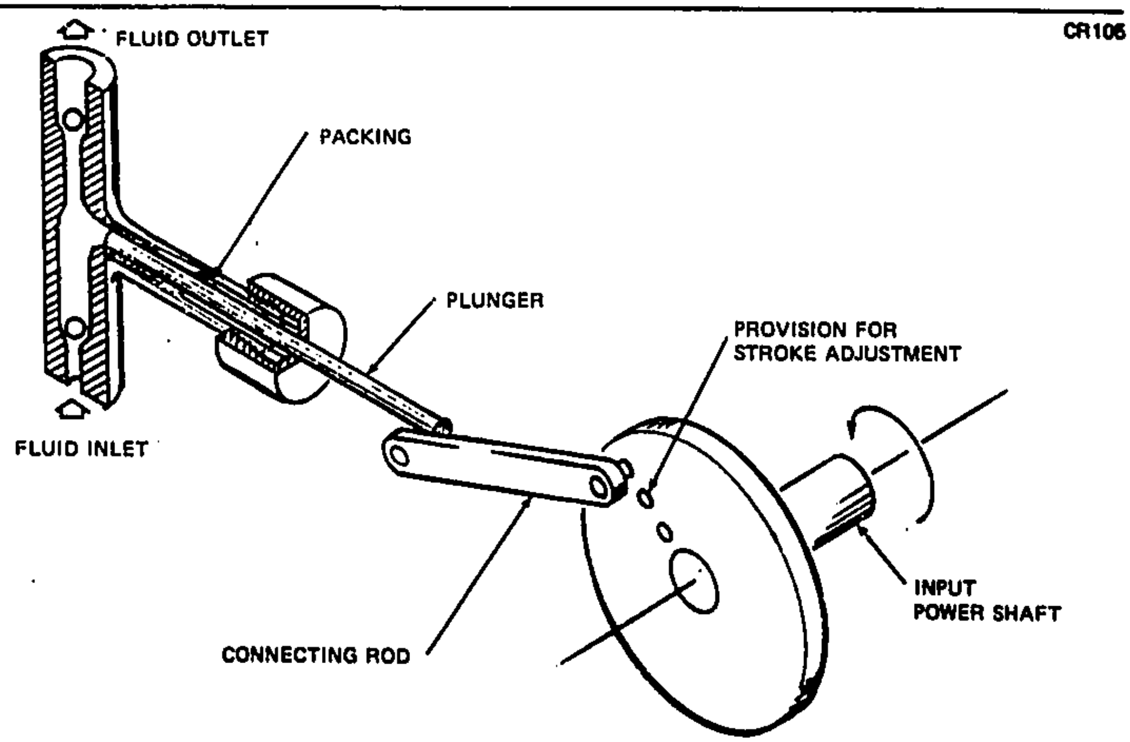

Figure 3-1. Packed Plunger Reciprocating Piston Pump

reciprocating and rotating parts and of materials compatibility are more easily solved.

Due to the cylinder and crankshaft arrangement and the supporting structure required, together with the fact that most designs employ speed reducers between the driving motor and input shaft, most crankshaft-driven reciprocating pumps are relatively heavy and have larger envelope volumes than other types of reciprocating pumps. However, several pumps of this type satisfy laboratory RO requirements as proper designs are available for high pressure water service, materials problems are minimized by use of diaphragms, and reliability is high.

3.1.1.2 Axial-Piston Pumps

Two principal types of axial-piston pumps are commonly used in aircraft and missile applications. In-line-type axial piston pumps (Figure 3-2) have the cylinder axial centerlines positioned parallel to the axial centerline of the power input shaft. The rotary motion of the input shaft is converted to reciprocating motion to drive the pistons by a wobble plate (swash plate). In

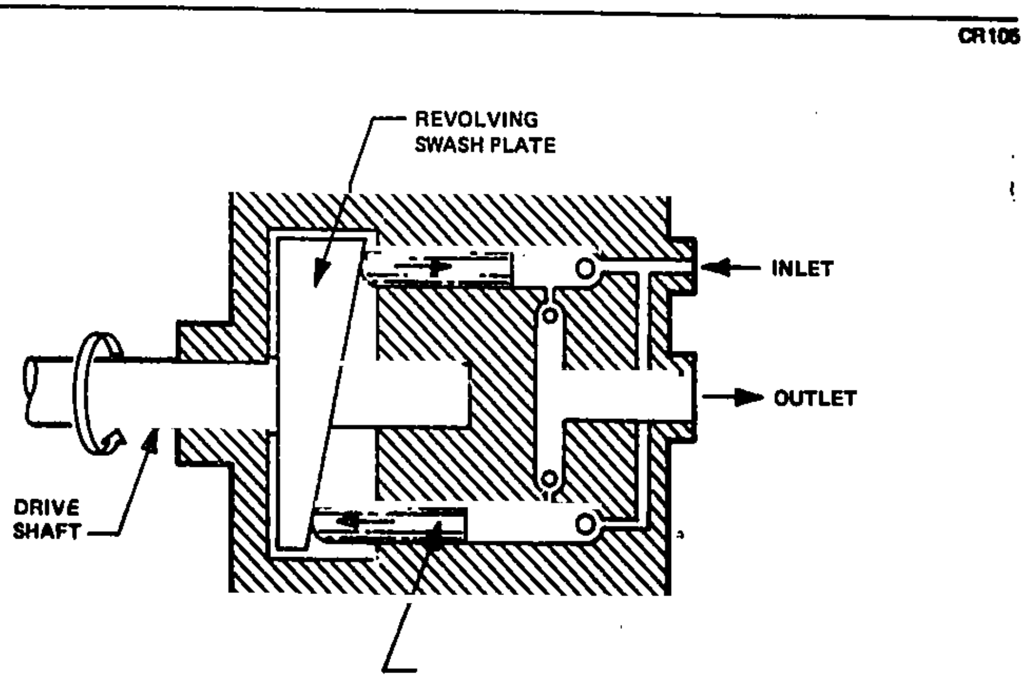

Figure 3-2. In-Line, Check-Valve, Axial-Piston Pump

bent-axis type axial piston designs, the cylinder assembly is placed at an angle to the drive shaft axis and allowed to rotate with the drive shaft (see Figure 3-3).

Check valves are used in some pumps to permit only unidirectional flow to the cylinder inlet and outlet ports. Check valves provide good sealing, are rugged, and tolerate shock and vibration well. Rotary valve plates may also be used to regulate the fluid flow.

Most aircraft pumps are axial-piston units of either the bent axis or in-line design. Bent-axis pumps are normally slightly more efficient in operation, but in-line designs are usually lighter and slightly less expensive.

Aircraft pumps are usually required to operate for the period between standard engine overhauls, usually 5,000 to 19,000 hours.

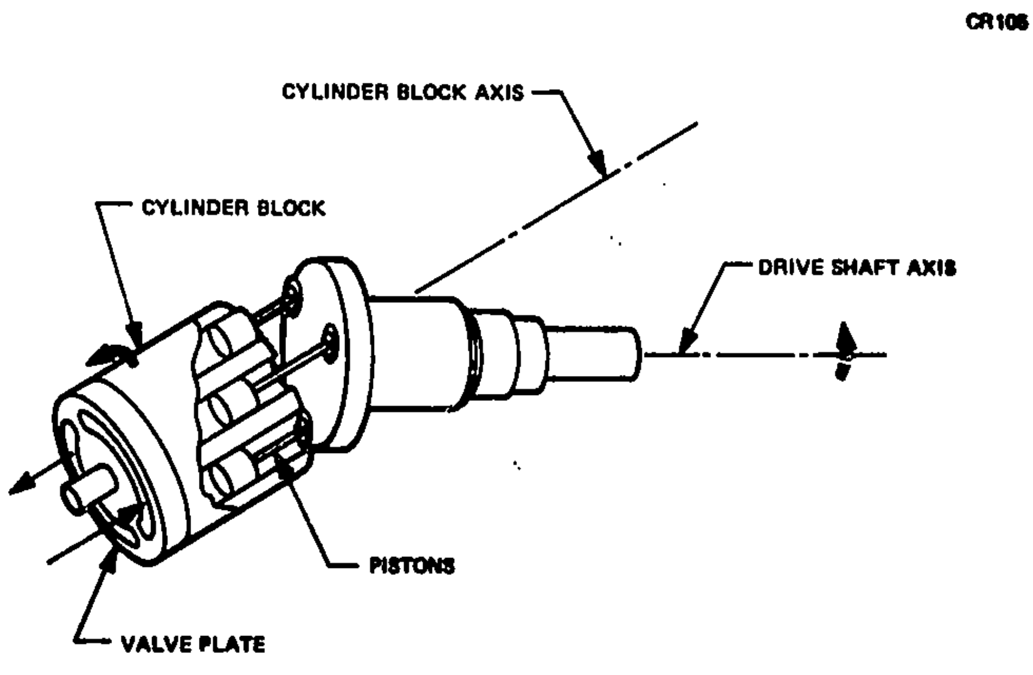

Figure 3-3. Bent-Axis, Valve-Plate Axial-Piston Pump

3.1.1.3 Radial Piston Pumps

In a radial piston pump the pistons are arranged radially, with their cylinder long axes in a plane perpendicular to the drive shaft axis. Radial piston pumps are available in both fixed and variable delivery types.

There are two commonly used types of radial piston pumps. Check-valve radial piston pumps have pistons driven by a rotating cam and use inlet and outlet check valves to direct fluid flow (see Figure 3-4). Pintle valve pumps have pistons attached to an outer circular reaction ring, which is mounted eccentric to the cylinder block. The reaction ring and cylinder block rotate together around a stationary pintle (see Figure 3-5).

Both check-valve and pintle-valve pumps may be designed for fixed or variable flow.

3.1.2 Centrifugal Pumps

Centrifugal pumps are selected for most industrial liquid transfer jobs. They have the advantages of simplicity, compactness, economy, reliability

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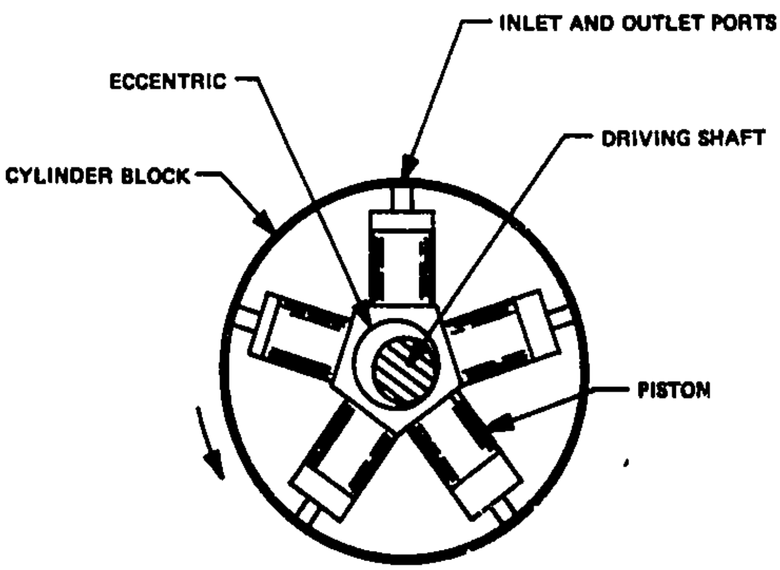

Figure 3-4. Radial-Piston Pump

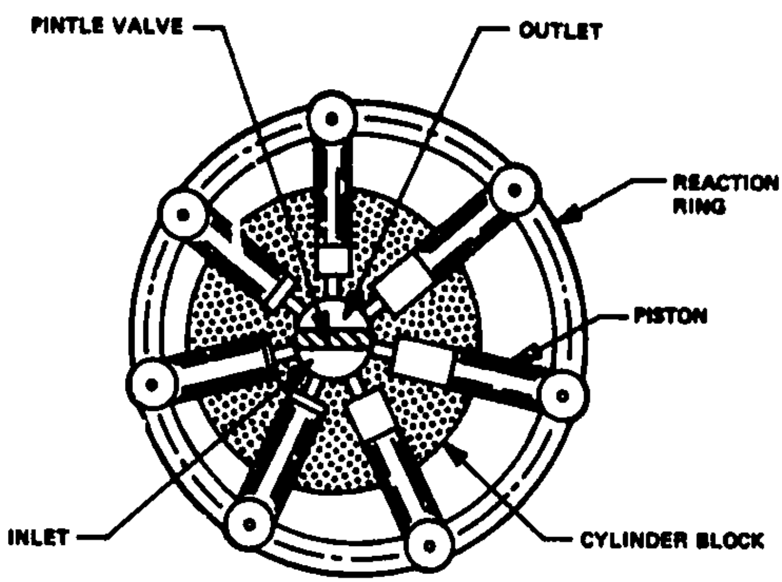

Figure 3-5. Pintle-Valve Type Radial-Piston Pump

and steady, nonpulsating flow. They are most efficient (up to 90 percent) in low viscosity (below 1,000 centipoises) constant flow (over 100 gpm), low to medium pressure (100 psi per pump stage) liquid transfer applications.

The specific speed, N_s, is a characteristic quantity of centrifugal pumps and provides an indication of the best type of impeller to use and the efficiency range to be expected. Specific speed, N_s, is expressed by the relation:

$$N_s = \frac{N\,Q^{1/2}}{H^{3/4}} \qquad (1)$$

where:

- N = impeller speed, rpm
- Q = flow rate, gpm
- H = pump output pressure, ft of head

Note that the specific speed is not dimensionless but includes a conversion factor involving g. If spacecraft RO pump conditions of N = 10,000 rpm, Q = 0.056 gpm and H = 1740 ft are assumed and these values are inserted in Equation (1), the resultant specific speed is 24.4. Centrifugal pump operation is, for most designs, most advantageous at specific speeds per stage of 500 to 15,000, and low specific speed is an indicator that pump efficiency will also be low. Thus centrifugal pumps are not well suited for the low-flow, high-pressure spacecraft RO application.

3.1.2.1 Volute, Diffuser, and Propeller Centrifugal Pumps

The most commonly used types of centrifugal pumps include: volute pumps (Figure 3-6), in which the liquid enters the impeller at its center and is accelerated outward into a channel of increasing area (volute) by centrifugal force imparted to it by the vanes; diffuser-type pumps (Figure 3-7), in which stationary diffuser blades in the casing around the circumference direct the output flow, and propeller pumps, which resemble a boat propeller encased in a tube. Mixed flow pumps are also sometimes used: these pumps provide

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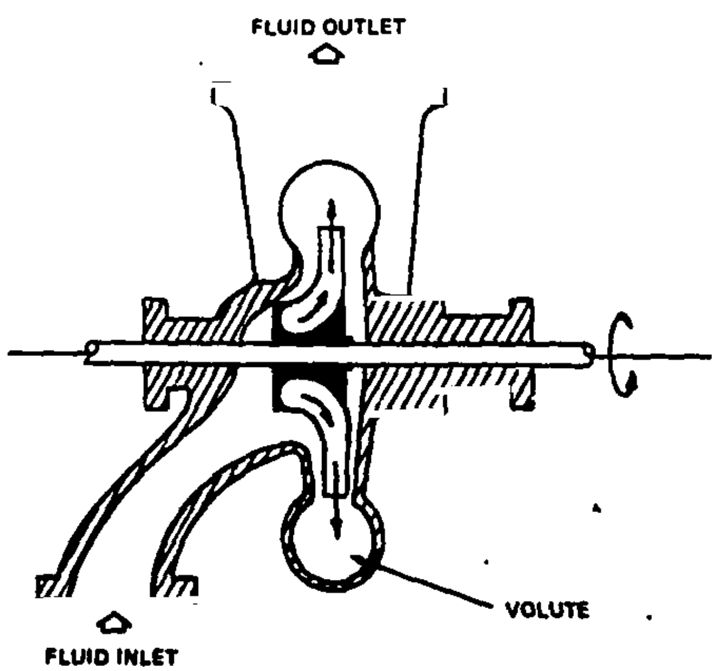

Figure 3-6. Volute-Type Centrifugal Pump

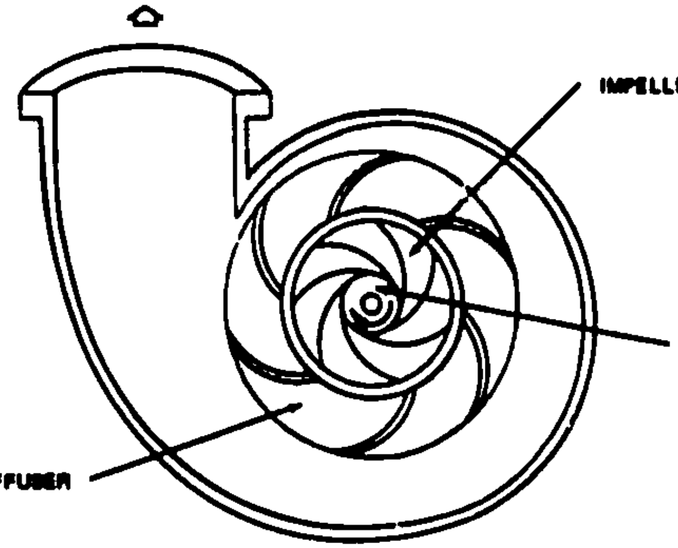

Figure 3-7. Diffuser-Type Centrifugal Pump

output pressure partly by propeller action and partly by centrifugal force developed in a volute casing.

3.1.2.2 Peripheral Centrifugal Pumps

The peripheral pump has a circular rotating impeller somewhat resembling a turbine wheel and is sometimes called a turbine pump. Peripheral pumps are low volume, high head pumps and have characteristics similar to those of positive displacement pumps (see Figure 3-8).

3.1.2.3 Impellers, Multiple Stages, and Centrifugal Pump Design

Impellers for centrifugal pumps may be open (blades attached to hub), semi-open (blades attached perpendicular to a disk), or closed (blades attached between a disk and a circular shroud). Open and semi-open impellers depend on close tolerances between the casings and the impellers to achieve high efficiencies. Closed impellers are more costly to manufacture but are not so susceptible to performance degradation due to wear.

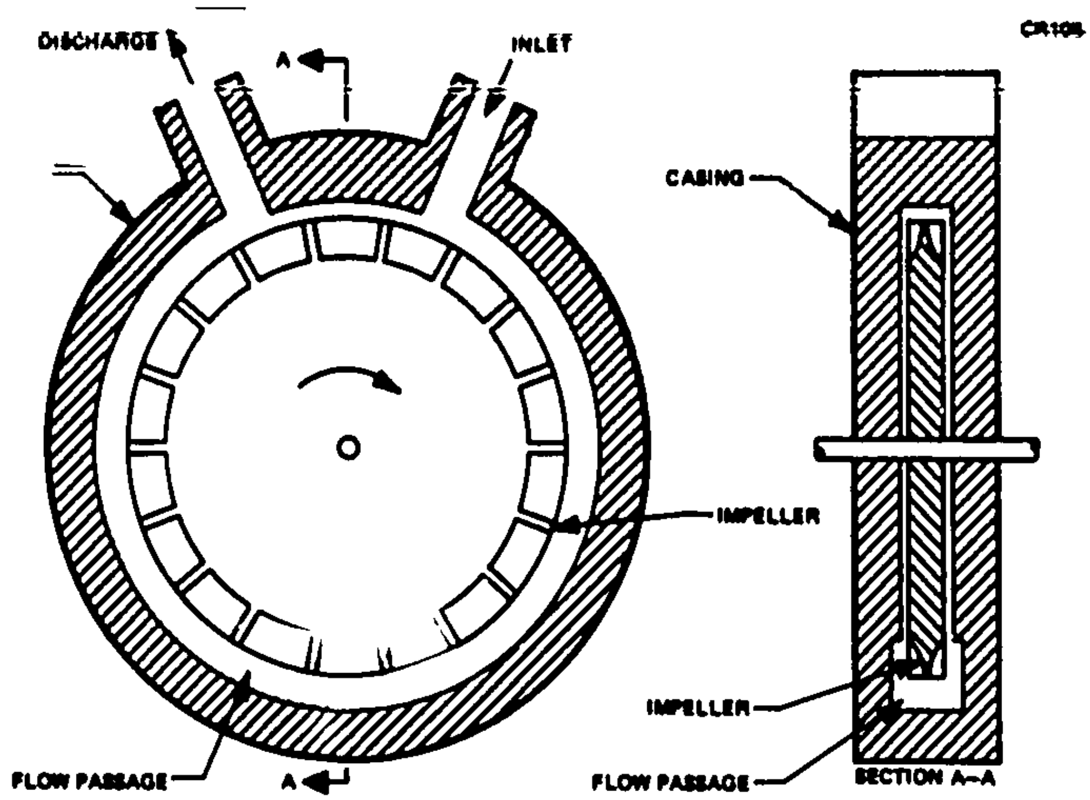

Figure 3-8. Peripheral Pump

Two or more centrifugal pumps may be combined on one shaft to produce increased output pressures. Due to the high-veloci... heads between stages, multiple-stage centrifugal pumps are more efficient than multiple single-stage centrifugal pumps connected in series. A typical multistage centrifugal pump is shown in Figure 3-9.

Pump manufacturers invest much time and money in optimizing centrifugal pumps for operating efficiency. New designs must be theoretically analyzed and then subjected to trial-and-error optimization test programs. Existing designs may be scaled up or down, however, using pump similarity relationships, and the efficiency of operation may be accurately estimated before fabrication is started.

3.1.3 Vane Pumps

A vane pump consists of a slotted rotor which turns in an eccentric housing. Figure 3-10 shows a typical vane pump configuration. Vanes inserted into the slots in the rotor maintain contact with the housing and provide positive

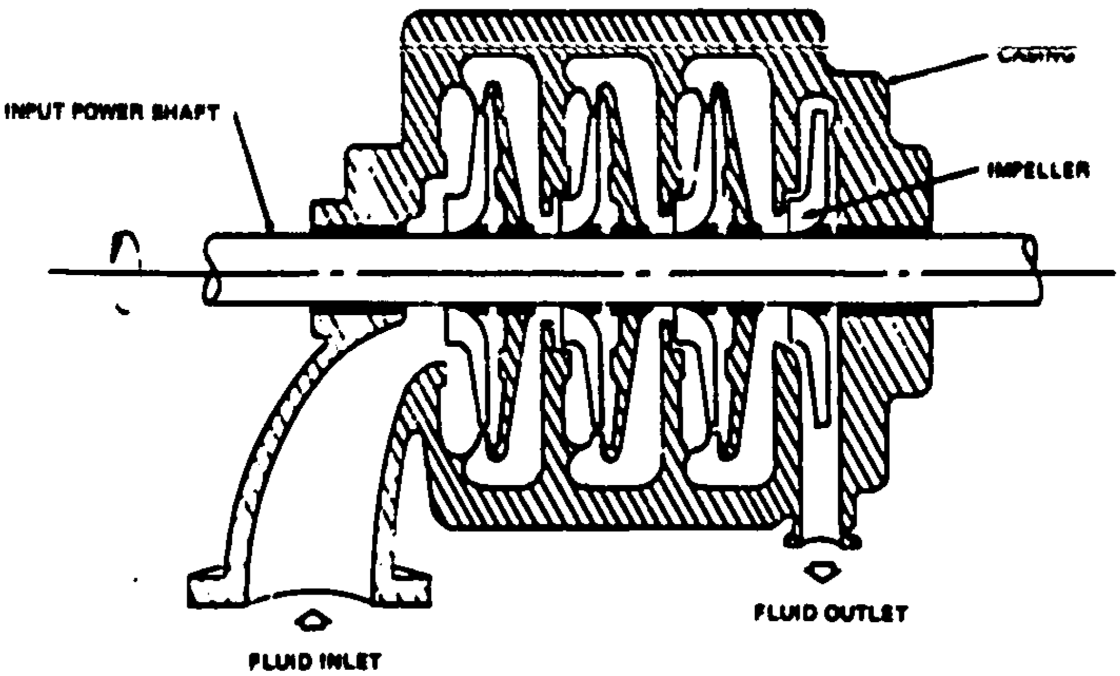

Figure 3-9. Multistage Centrifugal Pump

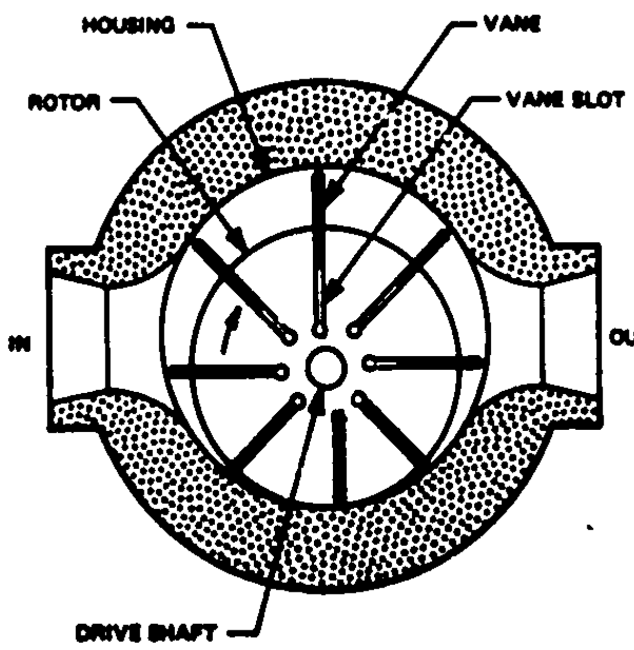

Figure 3-10. Vane-Pump

displacement pumping action. Passages machined into the sides of the housing provide passages for fluid inlet and outlet. Efficiencies of vane pumps are good, with fixed displacement vane pumps having efficiencies of 70 to 88 percent.

Fixed and variable displacement vane pumps are available. Variable displacement is achieved by varying the eccentricity of the housing to the rotor. This may be achieved on various models by manual, pneumatic or hydraulic means. Some variable displacement vane pumps are designed to provide automatic displacement variation by means of output pressure control of the housing eccentricity. Overall and volumentric efficiencies of variable displacement vane pumps are lower than for fixed capacity vane pumps, with overall efficiencies in the range of 70 to 80 percent.

Vanes are sealed against the housing either by centrifugal force, hydraulic pressure, spring loading, or a combination of methods. With proper vane design it is possible to combine low starting torque, low drag at working rpm, and low slippage.

Multiple vane pumps may be mounted on a single driving shaft and connected hydraulically either in series or parallel.

Vane pumps are easily maintained and some designs are provided with easily replaceable cartridges which contain all working parts. As the vanes wear they merely move radially outward, providing self-compensation for wear. Output of vane pumps is generally smoother than that of common gear or piston pumps.

In simple eccentric housing vane pumps the high pressure produced on the discharge side of the rotor can create quite high bearing and shaft loadings. Double eccentric or oval housing vane pumps have been produced which eliminate this disadvantage, but their configuration does not permit the incorporation of variable displacement features.

3.1.4 Gear Pumps

Gear pumps are widely used in hydraulic systems. They are low cost, positive displacement pumps with few moving parts, and are usually replaced on failure rather than rebuilt. (Figure 3-11 illustrates a typical gear-on-gear pump.) With proper inlet conditions gear pumps of less than 1 gpm capacity have been designed to run at up to 10,000 rpm. Gear pumps require careful design and precision manufacture to maintain the close internal clearances required to hold internal leakage to the low levels required for efficient operation. Bearing loads may be quite high resulting in high friction losses.

The three commonly used types of gear pumps are gear-on-gear, gear-in-gear, and axial flow.

3.1.4.1 Gear-on-Gear

Gear-on-gear pumps carry the working fluid from inlet port to outlet port between gear teeth and housing. The fluid being pumped may exert high forces on gear teeth, bearings, and housing; and gears must be precision machined and

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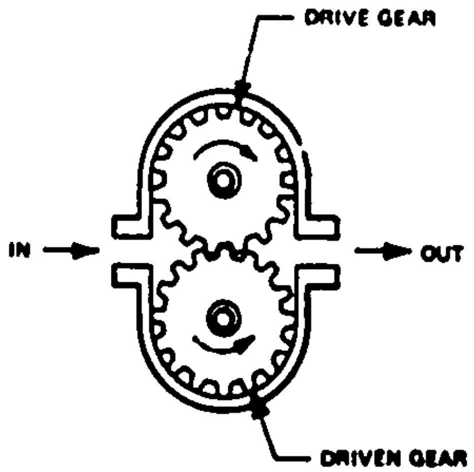

Figure 3-11. Typical Gear Pump

hardened. Most gear-on-gear pumps use two equal size gears. Spur gear pumps use straight-cut gears with large tooth depths and hence few teeth per gear. Tooth forms are modified to increase pumping efficiency and reduce noise.

Helical gear pumps operate more smoothly than spur gear pumps and produce an output with reduced pressure pulsations. Offsetting these advantages are the increased cost of the helical gears and the fact that end thrust is developed by the gears.

Three-gear pumps are available which provide almost twice the flow of conventional two-gear gear-on-gear pumps with a small weight and volume increase.

3.1.4.2 Gear-in-Gear Pumps

Gear-in-gear pumps usually employ straight out gears with modified tooth forms. In crescent-gear pumps a small inner gear runs in the internal teeth of a larger outer gear. Both gears turn in the same direction and fluid is carried from the inlet port to the outlet port sealed between the tips of the gear teeth and a stationary crescent shaped separator (Figure 3-12). The crescent gear pump is quiet in operation but is usually of low efficiency due to the difficulty in maintaining the close gear-tooth-to-crescent tolerances required.

The Gerotor pump is a gear-in-gear pump which is designed to eliminate the requirement for a crescent spacer. In this design the inner gear has one fewer tooth than the outer gear, so that with the teeth in mesh on one side of the outer gear, the tooth tips of the two gears just clear on the opposite side (Figure 3-13).

3.1.4.3 Axial Flow Gear Pump

An axial flow pump consists of a housing containing a pair of rotating meshing rotating screws. Usually called screw pumps, these pumps provide quiet, nonpulsating flow and are very quiet in operation. Several unfavorable characteristics limit their use: the contact conditions are unfavorable for sealing, the screws must be of at least one full turn in length, the casing

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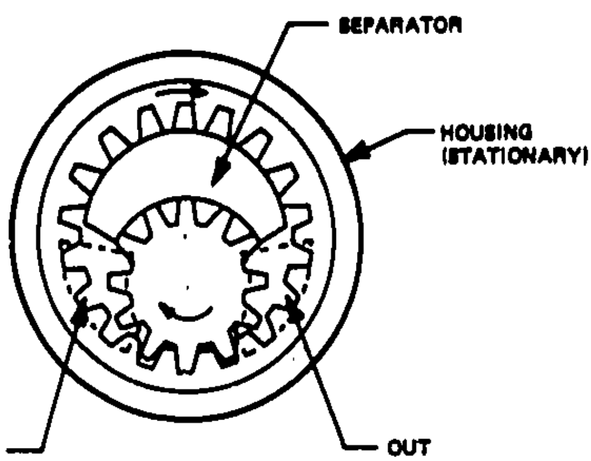

Figure 3-12. Gear-In-Gear Pump (With Separator)

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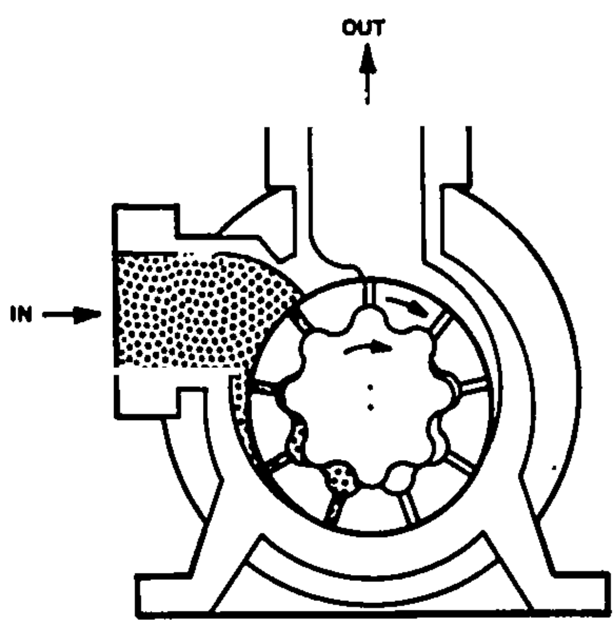


Figure 3-13. Gear-In-Gear Pump (Without Separator)

must be a precision fit around the gears, and the gears generate high thrust loads. Also, screw pump efficiencies are usually low compared to other positive displacement pumps. Figure 3-14 illustrates the principle of operation of a typical screw pump.

3.2 RESULTS OF PUMP AVAILABILITY SURVEY

A survey was conducted to compile a list of vendors who had off-the-shelf pumps of the type required for RO use. This study was carried out by reviewing MDAC vendor information files (which contain up-to-date microfilm records of most vendor catalogs), by interviewing personnel presently engaged in RO research at seven different companies and government organizations, and by reviewing vendor advertisements in engineering and research and development publications.

Using preliminary RO unit design data, compiled in the early stages of the RO unit definition study portion of this contract, preliminary RO pump requirements were formulated. A form letter was composed for transmittal to the

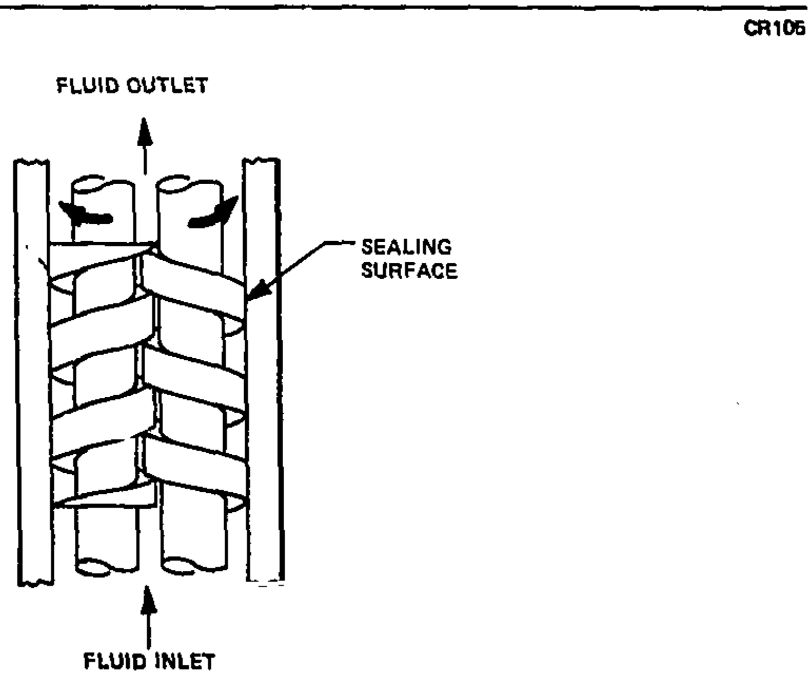


Figure 3-14. Two-Rotor Screw Pump

potential vendors, outlining the application of the desired pump (see Appendix A). A copy of this letter was sent to the 64 pump manufacturers and sales representatives listed in Appendix B. Most pump manufacturers had several lines of pumps and most sales representatives represented more than one manufacturer, thus the total number of pumps considered was greater than 100.

A total of 48 responses to the form letter were received; ten vendors responded positively with information on 15 recommended pumps. Of the 15 recommended pumps, 7 were eliminated as grossly unsuited for the intended application on the basis of the information submitted with the vendor responses. The remaining pumps were evaluated by the procedure outlined in Section 3.3 of this report. Most available pumps meeting the requirements specified were designed as injection pumps for use in the chemical, petroleum, or water utility fields. As a consequence they tended to be designed for reliable, long-life operation using commercial alternating current motors and simple proven mechanisms, but having relatively large weights and envelope volumes.

3.3 PUMP SELECTION CRITERIA

The selection criteria and methods used for test pump selection are outlined in the following sections.

3.3.1 Method

In order to select the pumps to be tested a matrix of selection criteria was developed and weighted to provide numerical comparison. Three members of the MDAC Biotechnology Department engineering staff with a cumulative total of over 40 years of pertinent mechanical engineering experience were selected to review the technical data sheets provided by the pump vendors and make ratings of the candidate pumps. Rating categories included not only items covering the suitability of the pumps for prototype laboratory RO systems, but items which evaluated the design of the pumps for suitability to spacecraft applications.

3.3.2 List of Rating Categories and Weighting Factors

A list of categories in which the candidate pumps were rated is shown in Table 3-1. From zero to 10 points were assigned in each of the 20 categories for each pump by the evaluators. Ten points represented the best score which could be assigned in any category. To insure that the rating categories deemed most important in the selection of pumps for the laboratory pump evaluation test had the most influence on the selection process, weighting factors were assigned each category. The value of the weighting factors ranged from 2 to 10, with 10 representing the category thought most important in the selection. Weighting factors are also listed in Table 3-1.

The number of points awarded to each pump in each category by the evaluators was multiplied by the appropriate weighting factor to produce an adjusted point score in each of the 20 categories for the individual pumps. The adjusted point scores in the 20 categories were then summed to obtain an overall adjusted point score for each pump.

3.3.3 Results of Pump Ranking

A summary of the results of the pump ranking is presented in Table 3-2. In addition to the overall adjusted point total awarded to each pump, the pump type, power requirements, envelope size (smallest right rectangular prism with which the pump and motor may be enclosed), and weight are presented in Table 3-2. The detailed pump rating results are presented in Table 3-3.

3.4 PUMP PROCUREMENT

Based on the results of the pump ranking study, the four highest rated pumps (Precision Control Products 1971-121, Milton Roy FR141A-72, Haskel MS-12, and BIF 1731-12-9511) were ordered for integration into the pump evaluation test stand. The selected test pumps are shown in Figure 3-15. A brief description of the features of each of these pumps is given in the following paragraphs.

The Precision Control Products 1971-121 pump is an electric-motor-driven type which uses a mechanically driven Hypalon diaphragm to displace the fluid being pumped. Inlet and outlet stainless steel check valves with double Teflon O-ring seats provide directional control of the fluid flow. This unit has a dry

Table 3-1

RATING CATEGORIES AND WEIGHTING FACTORS

Item	Category	Weighting Factor
1.	Compatibility of Component Materials (10 compatible—0 unsuitable)	10
2.	Simplicity of Design (10 simple—0 complex)	9
3.	Amount and Simplicity of Support Equipment Required and Interface Simplicity (10 little, simple—0 much, complex)	8
4.	Weight (10 light—0 heavy)	8
5.	Envelope Volume (10 small—0 large)	7
6.	Estimated Maintainability (10 easily repaired—0 repaired with difficulty)	7
7.	Service Requirements (Spares and Expendables Required and Service Interval) (10 low, infrequent—0 high, frequent)	7
8.	Uniformity of Output Pressure (10 uniform—0 severe surges)	7
9.	Estimated Life (10 long—0 short)	7
10.	Adaptability to Spacecraft Use (10 easily adaptable—0 difficult to adapt)	6
11.	Compatibility with Basic Design of RO Unit (10 compatible—0 incompatible)	6
12.	Input Power Requirements (10 low—0 high)	5
13.	Efficiency at Maximum Rated Pressure (Overall) (10 high—0 low)	5
14.	Potential for Power Reduction (10 excellent—0 poor)	3
15.	Potential for Weight Reduction (10 excellent—0 poor)	3
16.	Potential for Size Reduction (10 excellent—0 poor)	2
17.	Potential for Quiet Operation (10 excellent—0 poor)	2
18.	Potential for Increased Flow and Pressure with Existing Design (10 excellent—0 poor)	2
19.	Availability (10 standard design—0 unproven special)	2
20.	Cost (10 low—0 high)	2

Table 3-2

SUMMARY RESULTS OF PUMP RANKING

Pump Part No.	Vendor	Pump Type	Power Requirements Watts	Envelope Size in.^{3} △	Weight lb △	Overall Adjusted Point Score (1000 max. possible)
1971-121	Precision Control Products Corp. Waltham, Mass.	Mechanically displaced diaphragm, electrically driven.	370 △	3060	47	717
FR141A-72	Milton Roy Co. St. Petersburg, Fla.	Hydraulic displaced diaphragm, electrically driven.	780 △	2641	117	698
MS-12	Haskel Engineering and Supply Co. Burbank, Calif.	Packed plunger, pneumatically driven.	205 △	123	9	663
1731-12-9511	BIF Division, General Signal Corp. Providence, R.I.	Hydraulic displaced sleeve, electrically driven.	980 △	6870	120	643
80-5	Teledyne-Sprague Engineering Gardena, Calif.	Packed plunger, pneumatically driven.	634 △	155	7	604
D1-67096-04	Hills-McCanna Division, Pennwalt Corp. Carpentersville, Ill.	Packed plunger, electrically driven.	530 △	2120	150	579
MAC-24	McFarland Industries, Inc. Houston, Texas	Packed plunger, pneumatically driven.	154 △	5590	84	562
MAC-P-10	McFarland Industries, Inc. Houston, Texas	Packed plunger, electrically driven.	280 △	6580	150	548

Notes

△ Based on motor size. Actual power consumption to be determined in pump evaluation tests.

△ Based on power required to compress air supplied for 0.056 gpm and 800 psi output. A 72 percent compression efficiency (adiabatic power to compressor shaft input) was assumed.

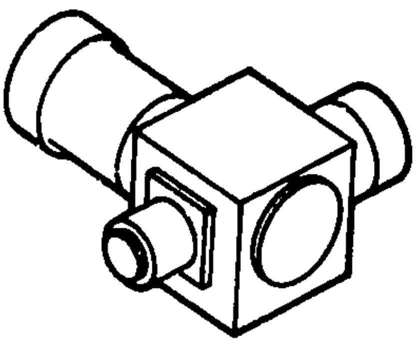
△ Volume of the smallest right rectangular prism which will enclose pump and motor. These volumes were used for comparison only. Actual volumes of selected pumps were computed after procurement (see text).

△ Includes motor.

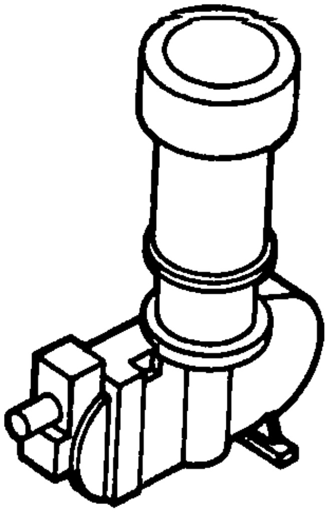
Table 3-3

DETAILED PUMP RATING RESULTS

Item No. of Rating Category (Ref. Table 3-1)	Weighting Factor	1971-121 Precision Points	Adjusted Point Score	FR141A-72 Milton Roy Co. Points	Adjusted Point Score	MS-12 Hackel Points	Adjusted Point Score	1731-12-9511 BIF Points	Adjusted Point Score	808 Teledyne-Sprague Points	Adjusted Point Score	D1-67096-04 Hills-McCanna Points	Adjusted Point Score	MAC-24 McFarland Ind. Points	Adjusted Point Score	MAC-R-10 McFarland Ind. Points	Adjusted Point Score
1	10	8	80	9	90	7	70	8	80	7	70	7	70	6	60	6	60
2	9	6	54	6	54	10	90	6	54	9	81	5	45	7	63	5	45
3	8	8	64	8	64	2	16	8	64	2	16	7	56	1	8	7	56
4	8	8	64	6	48	9	72	5	40	10	80	3	24	7	56	4	32
5	7	6	42	7	49	10	70	3	21	9	63	8	56	5	35	4	28
6	7	7	49	6	42	7	49	5	35	7	49	4	28	4	28	4	28
7	7	7	49	6	42	5	35	6	42	5	35	4	28	3	21	3	21
8	7	5	35	5	35	3	21	5	35	3	21	3	21	6	42	3	21
9	7	8	56	9	63	5	35	6	42	4	28	7	49	5	35	6	42
10	6	5	30	3	18	1	6	3	18	1	6	2	12	1	6	3	18
11	6	7	42	7	42	9	54	6	36	9	54	6	36	8	48	6	36
12	5	7	35	4	20	9	45	3	15	5	25	6	30	10	50	8	40
13	5	5	25	4	20	7	35	9	45	3	15	6	30	8	40	10	50
14	3	7	21	6	18	2	6	8	24	2	6	6	18	2	6	2	6
15	3	3	9	7	21	1	3	8	24	1	3	6	18	6	18	7	21
16	2	4	8	6	12	1	2	7	14	1	2	5	10	4	8	5	10
17	2	8	16	7	14	2	4	7	14	2	4	4	8	1	2	1	2
18	2	7	14	8	16	6	12	8	16	6	12	6	12	5	10	5	10
19	2	9	18	7	14	10	20	7	14	7	14	7	14	7	14	7	14
20	2	3	6	8	16	9	18	5	10	10	20	7	14	6	12	4	8
Overall Adjusted Point Score			717		698		663		643		604		579		562		548

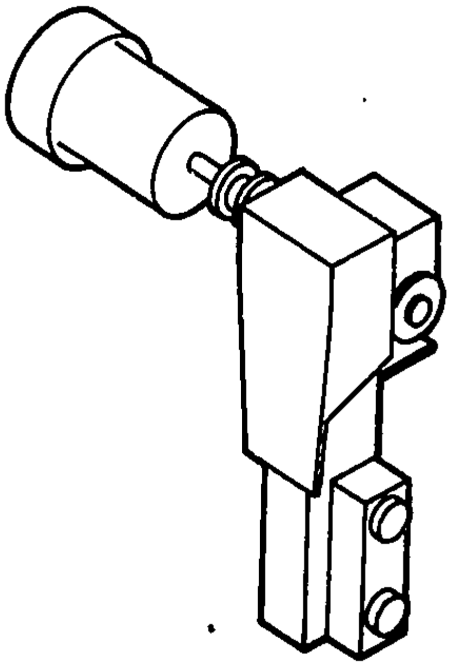


PRECISION CONTROL PRODUCTS 1971-121
VOLUME: 701 IN.^{3}
WEIGHT: 47 LB
MAX. RATED OUTPUT: 27 IN.^{3}/MIN AT 800 PSI
(1/10 SCALE)



MILTON ROY FR141A-21
VOLUME: 736 IN.^{3}
WEIGHT: 117 LB
MAX. RATED OUTPUT: 18 IN.^{3}/MIN AT 1800 PSI
(1/10 SCALE)

HASKEL MS-12
VOLUME: 86.4 IN.^{3}
WEIGHT: 9 LB
MAX. RATED OUTPUT: 43 IN.^{3}/MIN AT 1200 PSI
(1/10 SCALE)



BIF 1731-12-6611
VOLUME: 920 IN.^{3}
WEIGHT: 120 LB
MAX. RATED OUTPUT: 19 IN.^{3}/MIN AT 3000 PSI
(1/10 SCALE)

Figure 3-15. Test Pump Volumes, Weights, and Maximum Rated Outputs weight of 47 pounds and an actual volume of 701 in.^{3}. Speed reduction is accomplished by means of an oil bath waveform drive. This pump is of a recent design and is equipped with corrosion-resistant exterior components (with the exception of the drive motor). Pump output is not continuous and pressure pulsations require external suppression for RO applications.

The Milton Roy FR141A-72 uses a worm reduction gear driven eccentric to drive a hydraulic plunger. The hydraulic oil pressure developed actuates a flexible Teflon diaphragm to transmit energy to the pumped fluid. The reduction gears and mechanism are oil bath lubricated. Double ball check valves are used on the suction and discharge lines. An oil bypass port allows the adjustment of output flow volume while the pump is operating. Weight of the pump and motor combination is 117 lb, actual volume is 735 in.^{3}. Pump output pressure varies in a sinusoidal manner, and external pulsation dampening is required for RO application.

The Haskel MS-12 pump is an air driven pump which uses a large area air side piston to drive a smaller direct connected liquid end piston. The pump automatically reciprocates by the action of a pilot operated selector valve and return spring, and ceases pumping (with no air consumption) when the desired pressure has been reached. The air cylinder is impregnated with molybdenum disulphide and no air line lubrication is required. The pump has the disadvantages of requiring regulated compressed gas and of exhausting this gas at full input pressure. The pump is quite compact, weighing only 9 lb and occupying an actual volume of only 55.4 in.^{3}. Output pressure is easily controlled by varying input air supply, and the control system of an RO unit may be simplified with this type of pump. For RO use, output pressure surge control is required.

The BIF Model 1731-12-9511 pump has wetted parts constructed of 316 CRES, with a Hypalon diaphragm. The pump is electric motor driven through an oil bath lubricated worm gear speed reducer. Output of the speed reducer drives an eccentric which in turn drives a plunger which pressurises hydraulic fluid and expands the tubular diaphragm. The exterior of the tubular diaphragm is in contact with the fluid being pumped and as the diaphragm expands, the working fluid is forced through the outlet check valve. A slide valve in the hydraulic fluid chamber allows precise adjustment of pump stroke capacity. The BIF pump has a weight of 120 lb, and occupies an actual volume of 920 in.^{3}. Output pressure pulse suppression is required for RO use.

The Milton Roy and BIF pumps are, as a consequence of their original intended use, relatively heavy and have considerable bulk. The Precision Control Products pump is lighter and more compact, although its weight and bulk are much greater than comparable capacity pumps designed for aircraft and missile service. All three of these pumps meet the requirements for use in a small-capacity laboratory RO system, but all would require extensive redesign for space application. Such factors as size, weight, imput power type, and lubrication system design would require modification.

The Haskel pump is very compact and light, and its use allows RO unit control system simplicity. However, its use on a space mission is dependent on the inclusion in the vehicle of a pressurized gas system which could be used to power the pump. Although the electrical power cost to compress gas for use by this type of pump is considerably greater than the electrical power cost for a well designed electrically driven pump, the ease of control and simplicity of design of the pneumatic pump warrants an evaluation of its performance in this study.

Section 4
REVERSE OSMOSIS PUMP TEST

The four candidate RO pumps, which were selected for testing in Section 3, were integrated into a pump test stand and evaluated during a 30-day test. The pump test stand permitted the simultaneous testing of the four candidate RO pumps under simulated wash water reclamation system conditions. The test stand design incorporated automatic control and fail-safe features which permitted around-the-clock testing with minimum attention. The RO pumps and associated controls were contained in an 8 by 5-1/2 by 2-1/2 ft test console.

4.1 TEST STAND DESCRIPTION

The RO pump test stand is shown in Figures 4-1 and 4-2. In operation, wash water was supplied at approximately 165°F and 15 psig to each pump after being processed through a 0.45μ filter. The wash water then entered the four candidate pumps which were installed in parallel. The following nomenclature was established for each candidate pump: Milton Roy—P1; BIF—P2; PCP—P3; and Haskel—P4. The identification number used for a pump was also used throughout that pump's loop to identify its metering reservoir, flow orifice, accumulator, and flow control valve.

The three electrically driven pumps were controlled by pressure switches which permitted each pump to cycle in a preset pressure range. The Milton Roy and BIF pumps (P1 and P2) operated between 750 and 800 psig and the Precision Control Products pump (P3) operated between 550 and 600 psig. The air drive Haskel pump (P4) outlet water pressure was controlled to approximately 800 psig by the regulated plant air supply.

The design flow rate of 0.56 gpm for each pump was controlled by a fixed flow orifice and an adjustable metering valve connected in series. The orifice and metering valve were sized to permit maximum flows of 0.098 gpm

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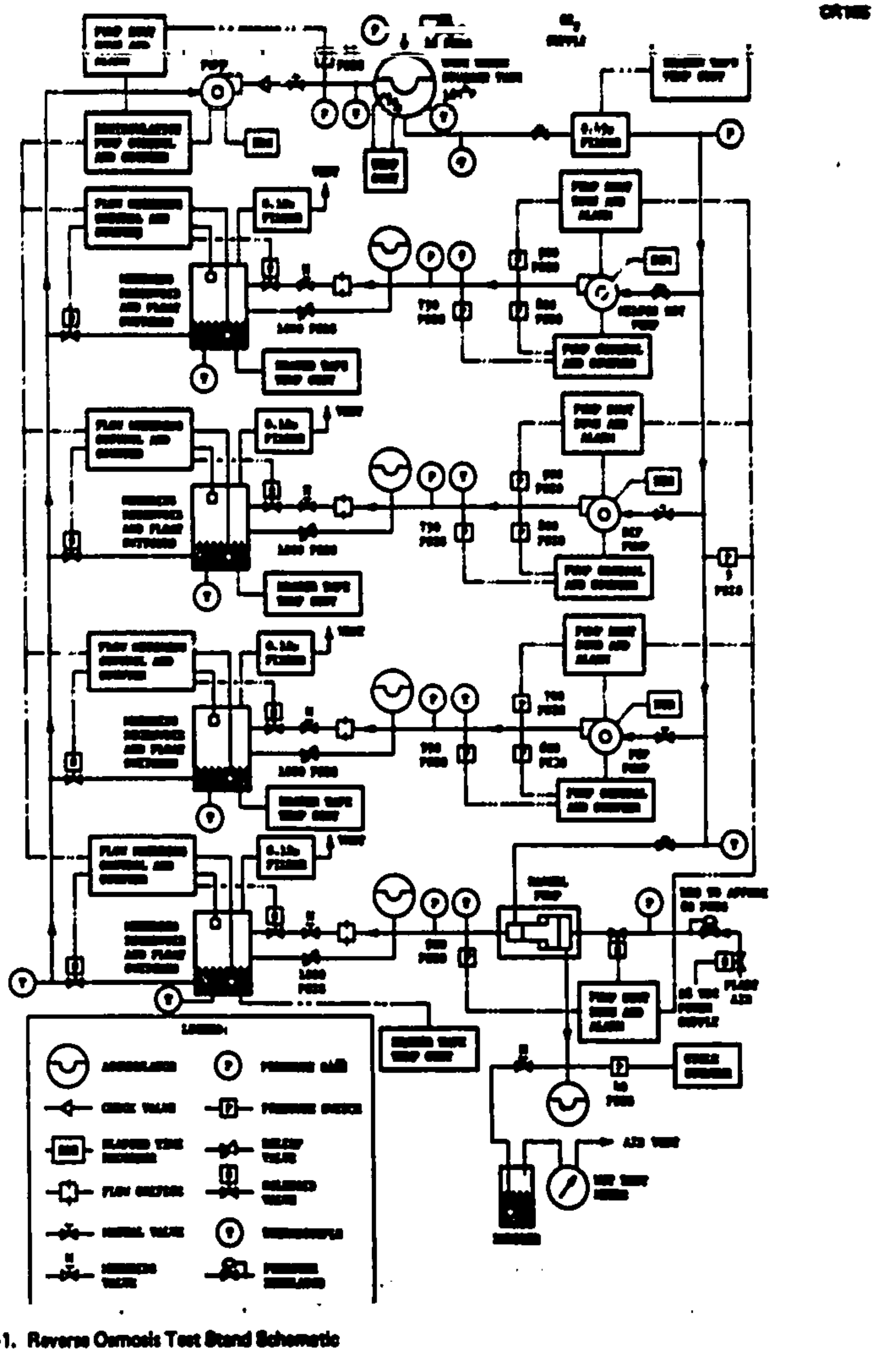

Figure 4-1. Reverse Osmosis Test Stand Schematic

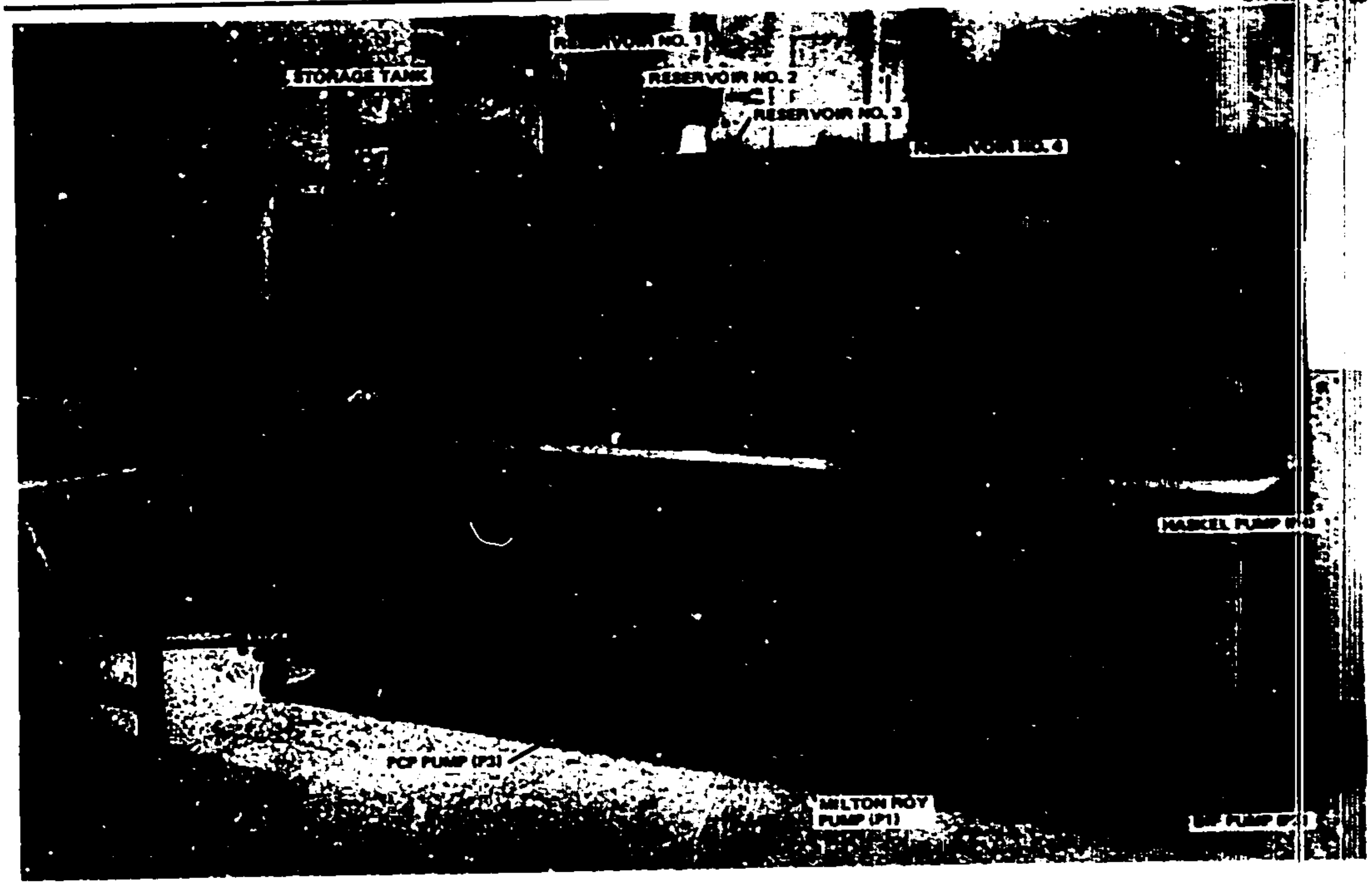

Figure 4-2. Reverse Osmosis Pump Test Stand During Test

at 800 psig and 0.085 gpm at 600 psig. The flow orifice was a Visco Jet, manufactured by the Lee Company, Westbrook, Connecticut. This unique orifice design provided the required flow resistance with a larger flow area than was possible with a single metering orifice, thereby minimizing the possibility of clogging the flow passage.

Transfer of the wash water back to the storage tank was accomplished by identical controls in each parallel test loop. After passing through the flow orifice and metering valve, the wash water flowed into a metering reservoir. Four metering reservoirs were provided, one for each pump being evaluated. When a reservoir was filled, the high float switch in that reservoir actuated to close the inlet solenoid valve, open the outlet solenoid valve, and start the recirculation pump, which transferred the wash water back to the storage tank from the reservoir. A low-level float switch was located in the reservoir at a calibrated increment of 1 gallon from the high float switch position. Actuation of the low-level float switch closed the outlet solenoid, deactivated the recirculation pump, and opened the inlet solenoid permitting the reservoir to be filled. The recirculation pump and the four metering reservoir controls were interlocked to permit only one metering reservoir pump-out cycle at a time. Thus each cycle represented the transfer of 1 gallon of water.

Automatic fail-safe features were provided to protect the test pumps. Overpressure protection was provided by high-pressure switches and backup relief valves in each test loop. An overpressure condition in a test loop shut down only the pump affected. the remaining three pumps continued operating.

An underpressure failure caused by loss of water actuated a low-pressure switch at the pump inlets, which shut off all pumps to prevent pump damage.

4.2 INSTRUMENTATION

The pump test stand was fully instrumented to provide data on performance parameters and power consumption of each test pump. The location of pressure and temperature instrumentation is shown in Figure 4-1.

Flow measurement was recorded for each pump by individual cycle counters which recorded the 1-gallon pumpout of each metering reservoir. The total flow for all four pumps was also recorded by a counter which totaled all recirculation pump cycles.

Average power consumption for the electrical pumps was determined by recording the total watt-hours and elapsed time of operation for each pump. The power consumption for the Haskel air-driven pump was determined by recording the total air usage, the inlet air pressure, and the pump duty cycle.

A list of the instrumentation for the test stand is shown in Table 4-1. The complete instrumentation and frequency of data recording is defined in the Test Plan (Appendix C).

Table 4-1

INSTRUMENTATION LIST

Parameter	Instrument	Quantity	Location
Temperature	Thermocouple	13	*
Pressure	Gages	8	*
RO pump flow	Cycle counter	4	Metering reservoirs
Total flow	Cycle counter	1	Recirculation pump
Pump power	Watt-hour meter	3	Electrically driven RO pumps
Heater power	Watt-hour meter	1	Storage tank
Duty cycle	Cycle counter	4	RO pumps
Operating time	Elapsed time recorder	4	Electrically driven RO pumps and recirculation pump
Air usage	Wet test meter	1	Haskel pump
Times of pump operation	Stripchart recorder	4	RO pumps
Times of reservoir operation	Stripchart recorder	4	Metering reservoirs

*See Figure 4-1

4.3 TEST PROCEDURE

After assembly of the test stand and the installation of the four candidate RO pumps, the test stand checkout and the pump test was conducted by the procedure outlined in the following paragraphs. The detail test procedure is described in the Test Plan (Appendix C).

4.3.1 Pretest Checkout

After assembly and electrical checkout, the test stand was filled with distilled water and operated at ambient temperature for 1 day. During this checkout the pump pressure control switches were adjusted to provide the desired operating pressure range and the reservoir float switches were set at approximately 1 gallon for each reservoir pumpout cycle. The actual calibration for each reservoir was:

Reservoir No. 1 (P1) = 0.972 gal/reservoir cycle
Reservoir No. 2 (P2) = 1.020 gal/reservoir cycle
Reservoir No. 3 (P3) = 1.001 gal/reservoir cycle
Reservoir No. 4 (P4) = 1.001 gal/reservoir cycle

After adjustments at ambient temperature, the storage tank, inlet filter, and reservoir heaters were activated to provide approximately 165°F at each pump inlet. The system performance was evaluated at temperature for an additional day. Prior to shutdown, a baseline water sample was taken from the system for chemical analysis.

The distilled water was drained from the test stand, the 0.45μ inlet filter element was replaced, and the system was refilled with wash water. After stabilizing at the test temperature and pressure conditions, the test stand was operated for 1 day to permit final adjustment and checkout of the test pumps, instrumentation, and the test stand peripheral equipment. Each test pump flow rate was adjusted to approximately the 0.056-gpm test flow requirement in accordance vendor information as follows:

Milton Roy (P1): Capacity Control = 72 percent of maximum
BIF (P2): Capacity Control = 68.33 percent of maximum
PCP (P3): Capacity Control = 48 percent of maximum
Haskel (P4): Adjust air pressure and flow control valve to obtain a pumping cycle rate of 35.8 strokes/minute.

Prior to the completion of the checkout test with wash water, a water sample was taken from the storage tank for chemical analysis.

4.3.2 30-Day Test

After completion of the system checkout, the initial load of wash water was drained from the system, the 0.45μ inlet filter element was replaced, and the system was refilled with 90.5 lb of simulated spacecraft wash water. The composition of this wash water is described in the Test Plan (Appendix C). In addition, the test plan describes the instrumentation, operating procedures, frequency of data collection, and chemical analysis required for this test.

Each of the four candidate RO pumps was scheduled to operate for 30 days (720 hours) under simulated spacecraft RO water recovery system conditions. Test time lost due to failures of peripheral test equipment was not counted as pump testing time and was made up. However, time lost due to pump failure was subtracted from the pump test period.

4.4 TEST RESULTS

The RO pump evaluation test was initiated at 1534 on April 6, 1973 and concluded at 1620 on May 8, 1973 for a total test duration of 768.8 hours. Of the four candidate RO pumps tested, only the Milton Roy (P1) and the BIF (P2) pumps successfully completed the required minimum of 720 hours of operation without pump malfunction. The PCP (P3) and Haskel (P4) pumps experienced many malfunctions and accumulated only 675.93 and 321.55 hours of normal operation, respectively. A detailed discussion of the performance of the four candidate RO pumps and the test stand is contained in the following paragraphs.

4.4.1 System Operation

A summary of the RO pump test operation is shown in Figure 4-3. The significant events which caused pump and/or test stand downtime are indicated by numbers in Figure 4-3 and are summarized in Table 4-2. A detail log of all significant events during the 30-day test is contained in Appendix D.

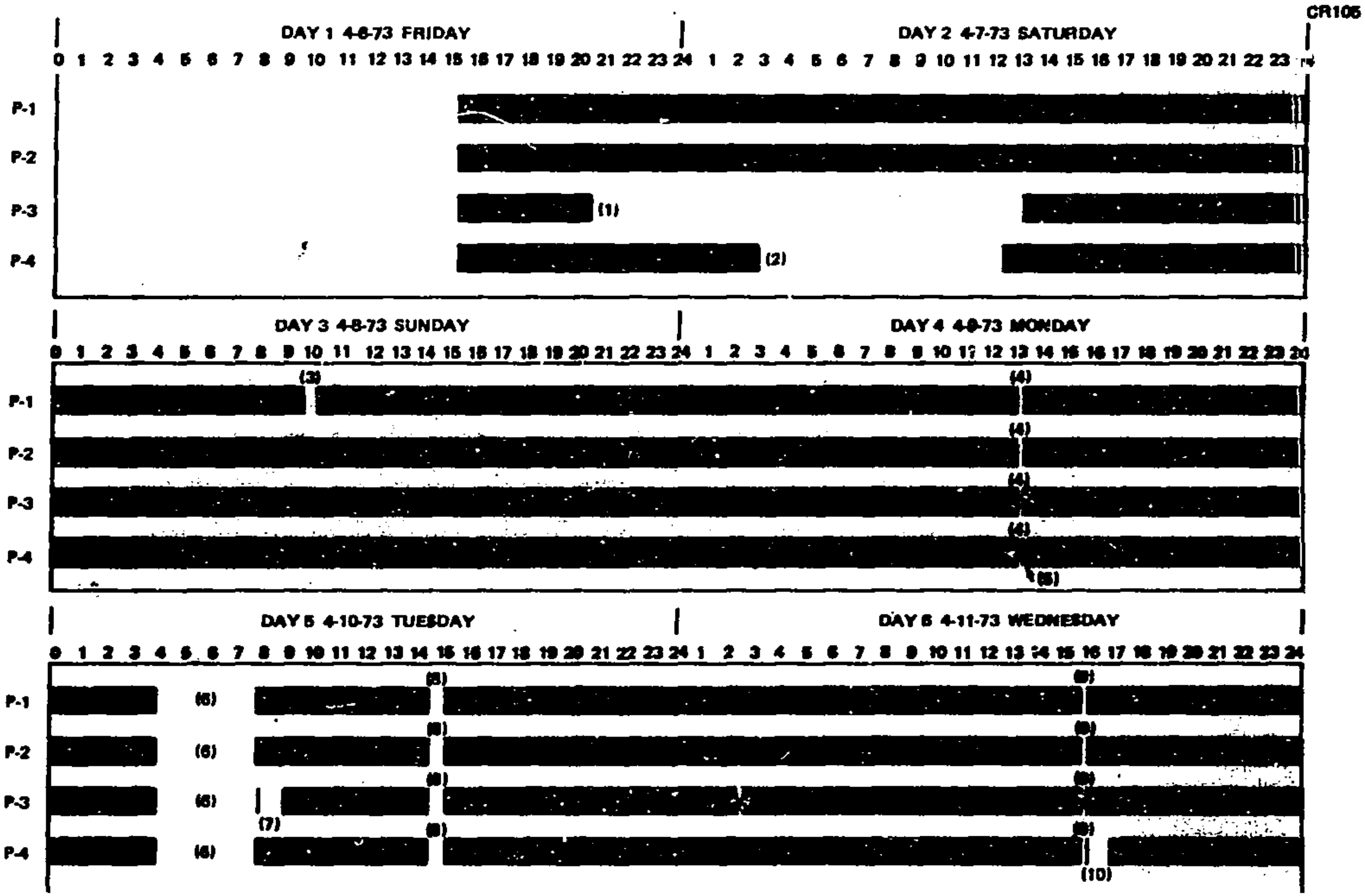

Figure 4-3. Summary of RO Pump Test Operation (Sheet 1 of 6)

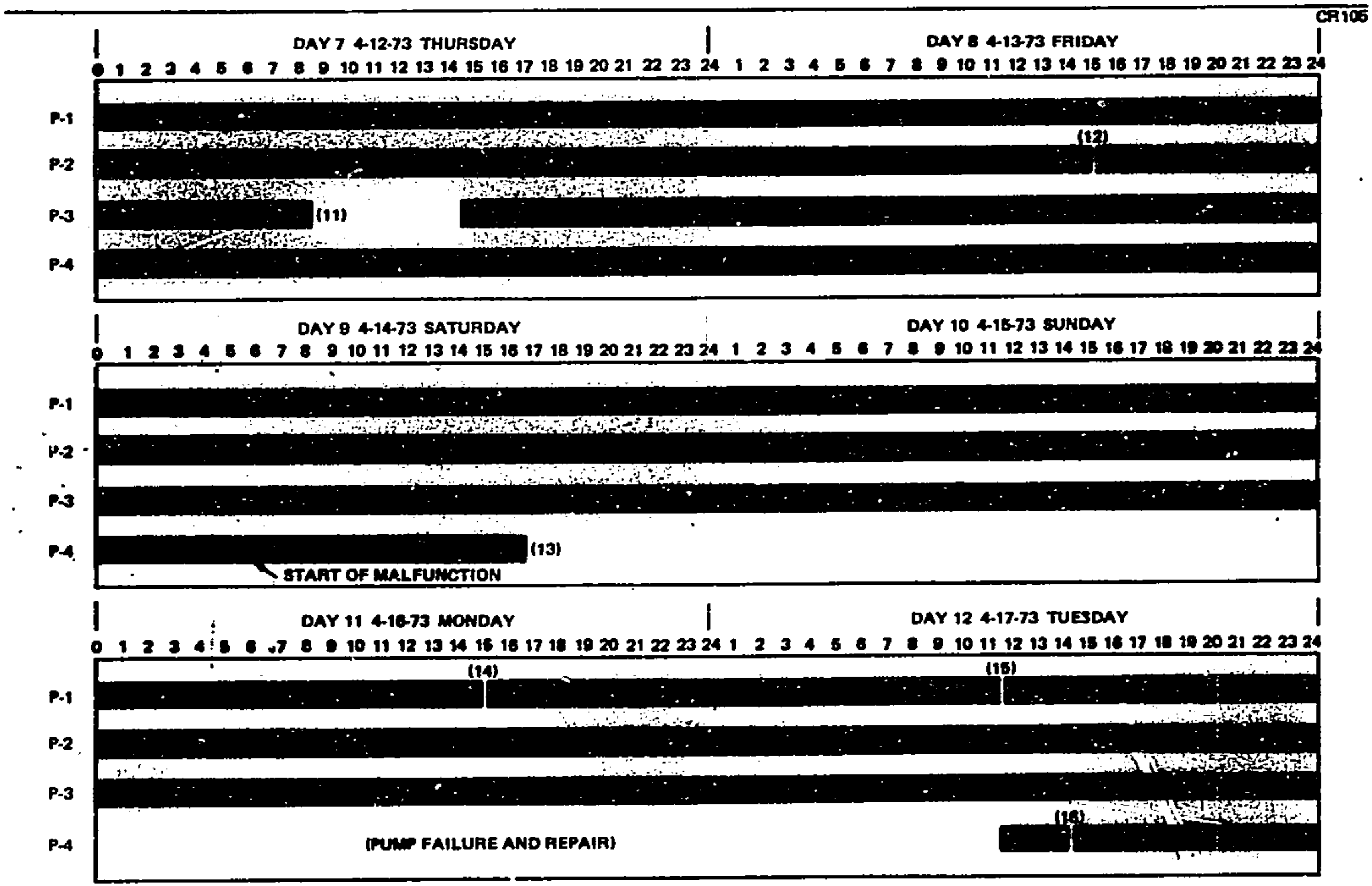

Figure 4-3. Summary of RO Pump Test Operation (Sheet 2 of 6)

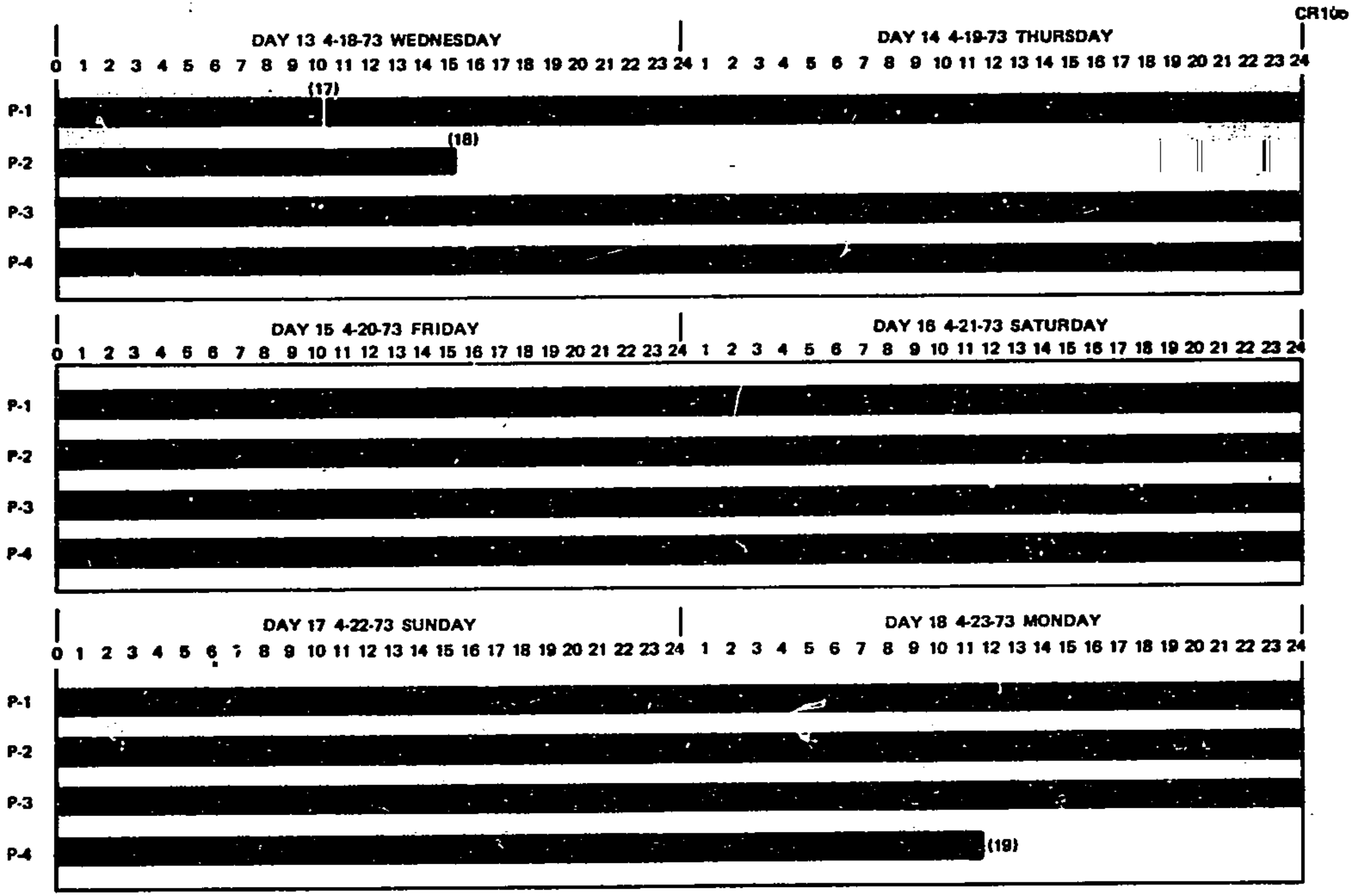

Figure 4-3. Summary of RO Pump Test Operation (Sheet 3 of 6)

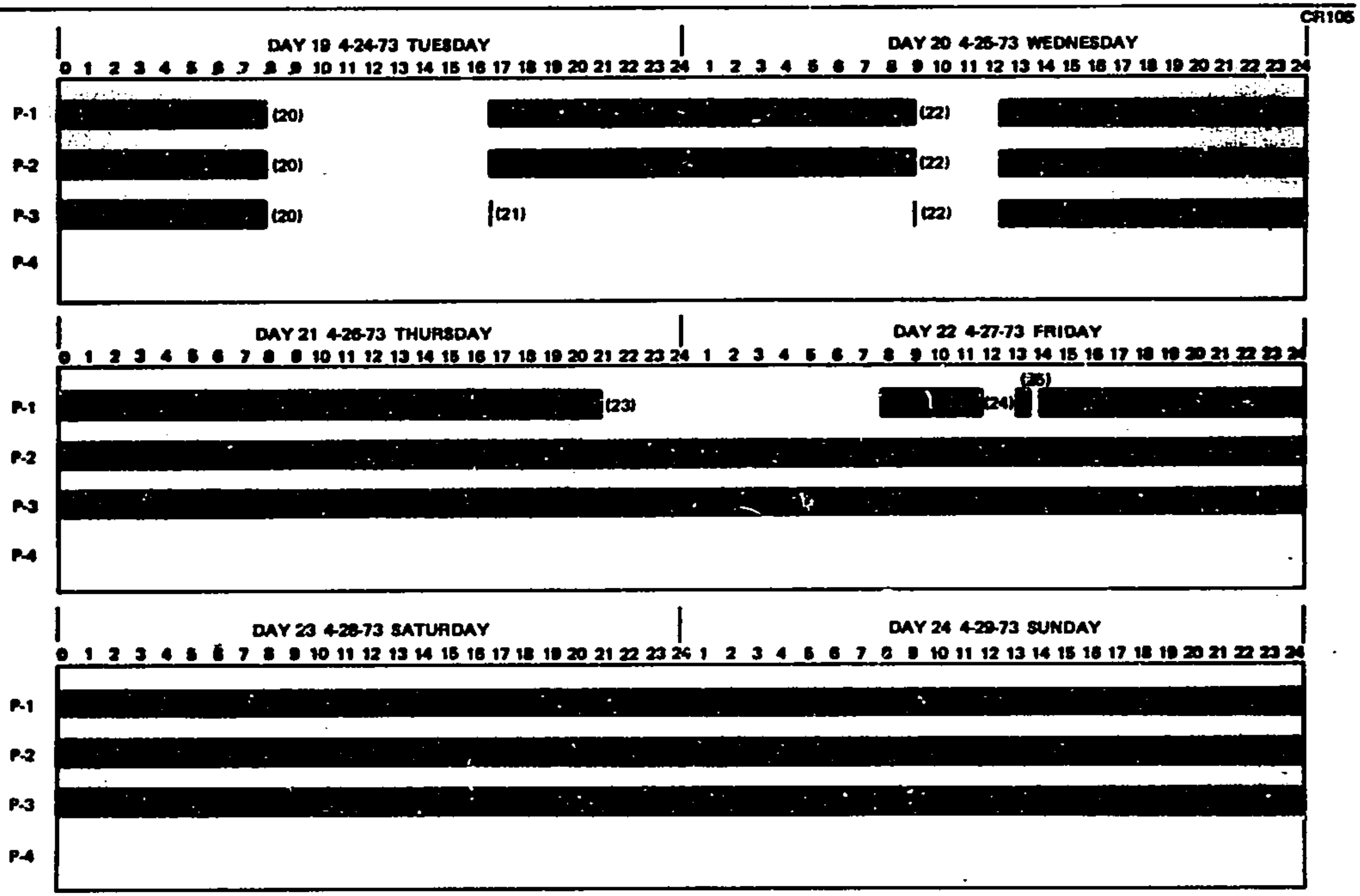

Figure 4-3. Summary of RO Pump Test Operation (Sheet 4 of 6)

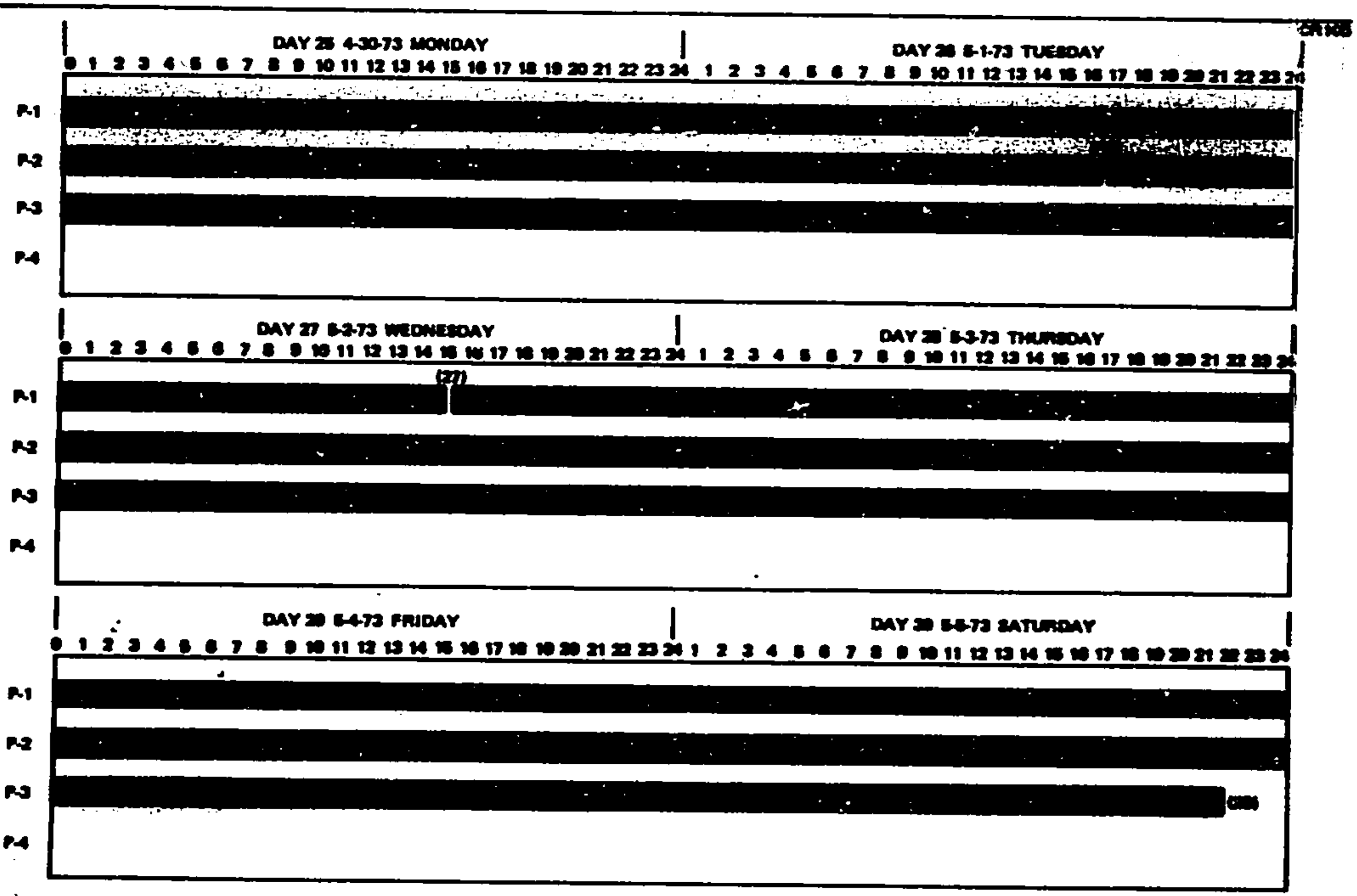

Figure 4-3. Summary of RO Pump Test Operation (Sheet 5 of 6)

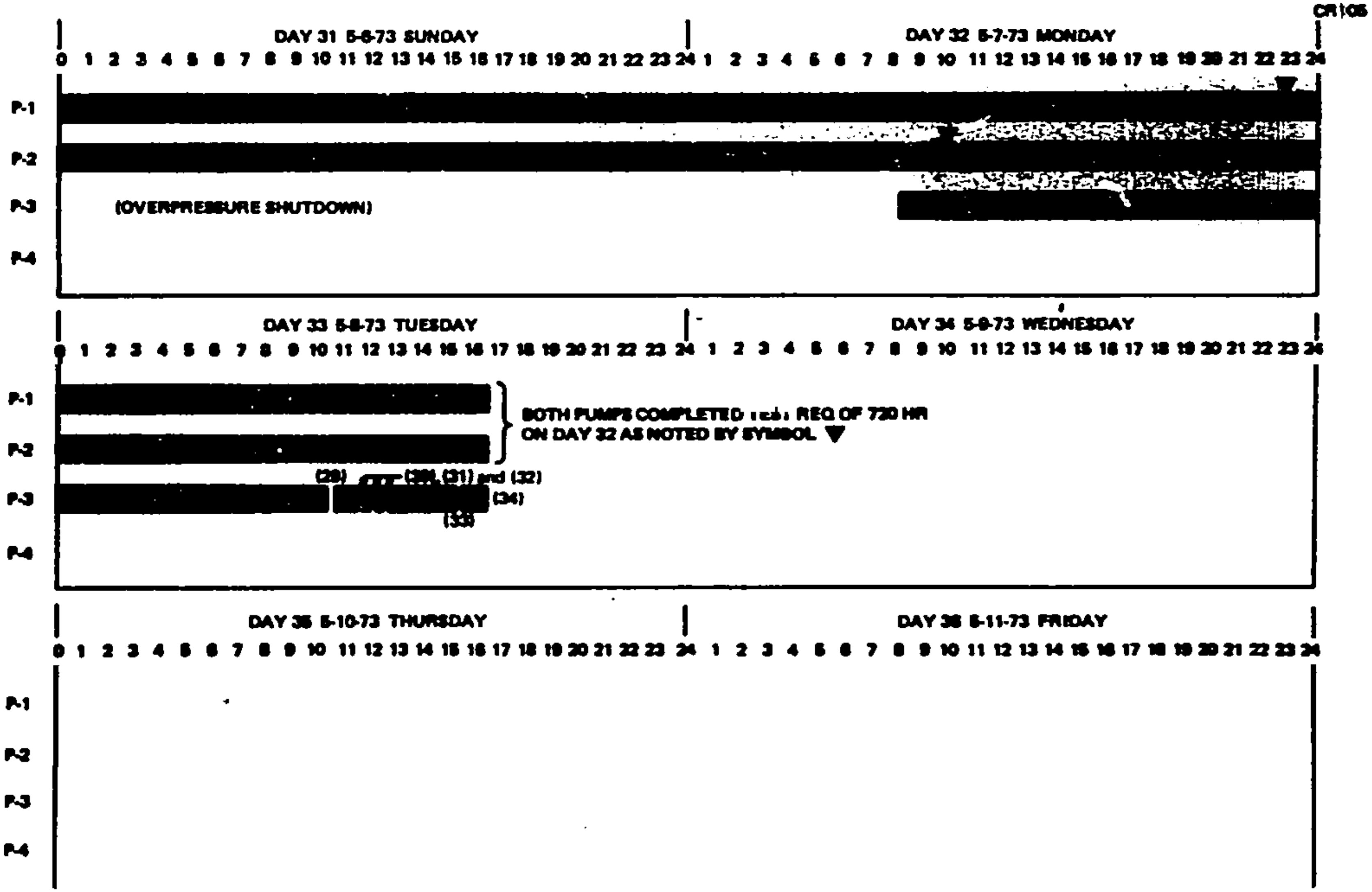

Figure 4-3. Summary of RO Pump Test Operation (Sheet 6 of 6)

Table 4-2

SUMMARY OF RO PUMP TEST SIGNIFICANT EVENTS

Event No. (See Figure 4-3)			
Pump Malfunction	Facility Malfunction	Pump and Facility Maintenance	Significant Event
(1)			P3 outlet valve O-ring failed on day 1 and was replaced on day 2.
	(2)		P4 shutdown on day 2 due to 900-psig overpressure condition with a facility air pressure of 85 psig. Facility air pressure was reduced to 80 psig which prevented overpressure shutdown, but the design water flow could not be achieved at this air pressure.
	(3)		P1 shutdown on day 3 due to a blown fuse which was replaced.
		(4)	Test stand was shutdown on day 4 to install a high-pressure control pressure switch for P4. This pressure switch limited the water outlet pressure to 840 psig with a facility air pressure greater than 80 psig.
(5)			P4 would not start normally after test stand shutdown and required a jolt from a hammer before starting.
	(6)		The RO pumps were shutdown on day 5 by a malfunction of a recirculation pump control relay. Problem corrected by adjustment of the power supply voltage.
(7)			P3 outlet valve O-ring failed again on day 5 and was replaced (Ref: Item No. 1).
	(8)		Test stand was shutdown on day 5 to replace a clogged Visco Jet and malfunctioning solenoid valve in the P1 system.
		(9)	Test stand was shutdown on day 6 to replace the inlet filter element.
(10)			P4 would not start normally after test stand shutdown and required a jolt from a a hammer before starting.
(11)			P3 outlet valve O-ring failed again on day 7 (Ref: Items No. 1 and 7). A smaller cross-sectional area O-ring was installed.

Table 4-2 (Page 2 of 3)

SUMMARY OF RO PUMP TEST SIGNIFICANT EVENTS

Event No. (See Figure 4-3)			
Pump Mal-function	Facility Mal-function	Pump and Facility Maintenance	Significant Event
	(12)		P2 shut down on day 8 to replace a clogged Visco Jet.
(13)			P4 failed on day 9. Pump was completely rebuilt and returned to operation on day 12.
		(14)	A pressure snubber was installed on the P1 system high-pressure control switch on day 11 to minimize pressure surges.
		(15)	The pressure snubber installed in (14) above was replaced on day 12 with a heavier unit to eliminate the surges.
	(16)		P4 pump cycle counter failed and was replaced on day 12.
	(17)		P1 shut down on day 13 to replace a clogged Visco Jet again (Ref: Item No. 8).
		(18)	P2 shutdown on day 13 to change oil at 240 hours per vendor's recommendation.
(19)			P4 failed on day 18. This failure was considered final.
	(20)		Test stand shutdown on day 19 due to a complete loss of water caused by a ruptured pressure switch in the P1 system. The failed pressure switch was replaced.
	(21)		P3 accumulator bladder was found to have failed after test stand restart. The failed polyurethane bladder was replaced with a Buna-N unit.
		(22)	Test stand shutdown to add additional water.
	(23)		P1 shutdown on day 21 due to 900-psig over-pressure condition. Determined to have been caused by drifting of setting on new pressure switch (installed on day 19) which was reset on day 22.
	(24)		P1 accumulator bladder was found to have failed on day 22. Since spares were not available, the accumulator from the P4 system was installed. The Visco Jet and

Table 4-2 (Page 3 of 3)

SUMMARY OF RO PUMP TEST SIGNIFICANT EVENTS

Event No. (See Figure 4-3)			
Pump Mal-function	Facility Mal-function	Pump and Facility Maintenance	Significant Event
			metering valve were found to be clogged with pieces of the failed bladder and were replaced.
	(25)		P1 pressure switch (Ref: Item No. 23) malfunctioned again and was replaced.
		(26)	P2 accumulator bladder and Visco Jet were replaced on day 26 as a precautionary measure. New natural rubber bladders had been obtained to replace the failing polyurethane bladders.
		(27)	P1 accumulator bladder (P4 accumulator installed on day 22) and Visco Jet were replaced on day 27 as a precautionary measure. A natural rubber bladder was installed.
	(28)		P3 shut down on day 30 due to 700-psig over-pressure condition. The high-pressure switch had drifted to 640 psig and was reset to 600 psig on day 32.
(29)			P3 shut down on day 33 due to 700-psig over-pressure condition. Since the pressure indication was only 600 psig, a pump pressure surge problem was suspected.
(30), (31) and (32)			P3 was shut down for inspection and evaluation three times in an attempt to determine the cause of the pressure surge.
		(33)	P3 was shut down to permit calibration of the outlet pressure transducer in order to record the suspected pressure surges on an oscilliscope.
(34)			P3 was shut down and considered failed when the oscilliscope traces revealed large pressure spikes (100 to 1,200 psig) at the outlet. Since P1 and P2 had completed the required 720 hours on day 32, the test was terminated.

9 Total maintenance items.
13 Total facility malfunctions.
12 Total pump malfunctions (8 on P3; 4 on P4).

As shown in Figure 4-3 and defined in Table 4-2, there were a total of 34 events which caused pump and/or test stand shutdown. Of those events: 9 were classified as pump and test stand maintenance items, which were required for the normal operation of the test; 13 were classified as facility malfunctions of peripheral equipment; and 12 were classified as actual RO pump malfunctions. Pump operation time lost due to maintenance and facility malfunctions was not counted as pump test time, and was made up, which accounted for the fact that the P1 and P2 pumps did not accumulate 720 hours until day 32. Operation time lost due to pump malfunctions was subtracted from the pump test period per the test plan. Therefore, P3 and P4 only accumulated 675.93 and 321.55 hours respectively prior to final failure. The P3 and P4 malfunctions are discussed in detail in Section 4.4.2. However, it should be noted that some of the long intervals of downtime due to either facility or pump malfunctions, noted in Figure 4-3, were because the test stand was designed for automatic shutdown in the event of a facility or pump malfunction and 24-hour engineering or technician coverage was not required. Whenever a failure occurred during an "off-shift" period, corrections were not initiated until the next scheduled observation period.

The majority of the facility malfunctions noted in Table 4-2 were minor and the automatic fail-safe features of the control prevented damage to the test pumps. The most significant facility malfunctions occurred on day 19 when a pressure switch failure caused the loss of all wash water from the stand. The automatic low-pressure shutdown control functioned correctly and prevented pump damage due to the loss of water at the pump inlets. After the pressure switch was replaced and the system refilled with water, it was discovered that the accumulator bladder for P3 had also failed. Investigation revealed that the rubber bladder was deteriorating and particles of rubber were clogging the Visco Jet flow orifices. The P3 system was cleaned, the Vico Jet replaced, and a spare accumulator installed.

Subsequent investigation of this bladder failure revealed that the four accumulators delivered for this test utilized bladders fabricated from polyurethane rubber which was recommended for the service by the manufacturer, Von Manufacturing Co., Inc., Los Angeles, California. This type of elastomer, sometimes referred to as urethane rather than polyurethane, is formed through the reaction product of a diisocyanate and a polyether or polyester glycol, identifying the urethane as either a polyester or polyether type. Both types generally exhibit the same physical properties.

A literature search on the suitability of urethane elastometers for water service revealed several inconsistencies. Reference 8 indicates that urethane elastometers exhibit little or no effect at 212°F when exposed to water. However, References 9 and 10 state that urethane elastomers are not recommended for water service. Nevertheless, the observed results of the test indicated that urethane rubber was not suitable and therefore all accumulator bladders should be replaced.

An attempt was made to contact the Von Manufacturing Co. to obtain additional information and a substitute bladder material. However, this company had experienced financial difficulties and after January 1973, when the four accumulators were delivered, they had gone out of business. After further investigations, a distributor was located who had handled Von accumulators. He reported that prior to 1972 the bladders had been constructed with Buna-N rubber which was satisfactory for water service. However, Buna-N bladders were no longer available except on special order from a local rubber product manufacturer. Fortunately the spare accumulator, installed in the P3 system on day 19, was manufactured in 1970 and did contain a Buna-N bladder.

Since Buna-N bladders could not be obtained in time to complete the test within schedule, a search was initiated to obtain an acceptable substitute for the remaining two accumulators. Bladder material of natural rubber was obtained and installed in the P2 and P1 accumulators on days 26 and 27 respectively. Since P4 had failed on day 18 ending its test, accumulator No. 4 was not repaired.

This facility malfunction illustrates the importance of system testing under actual design conditions, since vendor test data can be subject to incorrect interpretation. Although Reference 8 indicated that urethane elastomers were acceptable for water service at 212°F, the authors did caution the reader that the data were compiled from many sources and some data were not consistent due to the wide disparity in the types of test used by investigators and the lack of sufficient information on test conditions. Further, the authors of Reference 8 advised that, "If you're not familiar with the behavior of a material in a certain environment, run a laboratory test, then confirm it under service conditions. It could save you a lot of headaches later."

The results of the chemical analysis of the wash water, both for checkout and the 30-day test, are shown in Figures 4-4, 4-5, and 4-6 and in Table 4-3. The frequency of collection and the method of analysis is described in detail in the Test Plan (Appendix C).

In addition to the pH and conductivity analysis shown in Figures 4-4 and 4-5 for the distilled water checkout, the following baseline analysis was also performed:

TOC = 23 PPM
Petroleum Ether Extract = ~0.0003 percent

This additional analysis was performed to assure that hydraulic oil had not been introduced into the system by the pumps. These results indicated that a very clean system was achieved prior to the start of the test.

4.4.2 Pump Performance Evaluation

A summary of the temperature and pressure data obtained within the test stand during the test is shown in Figure 4-7. The data on this figure were obtained from all data recorded during the test and show that, except for a few minor exclusions, the pump test requirements were maintained in accordance with the test plan. The temperature requirement at the pump inlets was 165°F and the average temperature was 167.4°F. The P1, P2, and P4 pumps were required to operate between 750 and 800 psig and the P3 pump was required to operate at 550 to 600 psig. The average pressures were: 775 psig for P1; 773 psig for P2; 590 psig for P3; and 752 psig for P4.

The results of the RO pump test are summarized in Table 4-4. This summary includes all significant data recorded from the stripchart recorder, counters, elapsed time recorders, and watt-hour meters. From these data the pump flows, power requirements, and efficiencies were computed and are included in Table 4-4.

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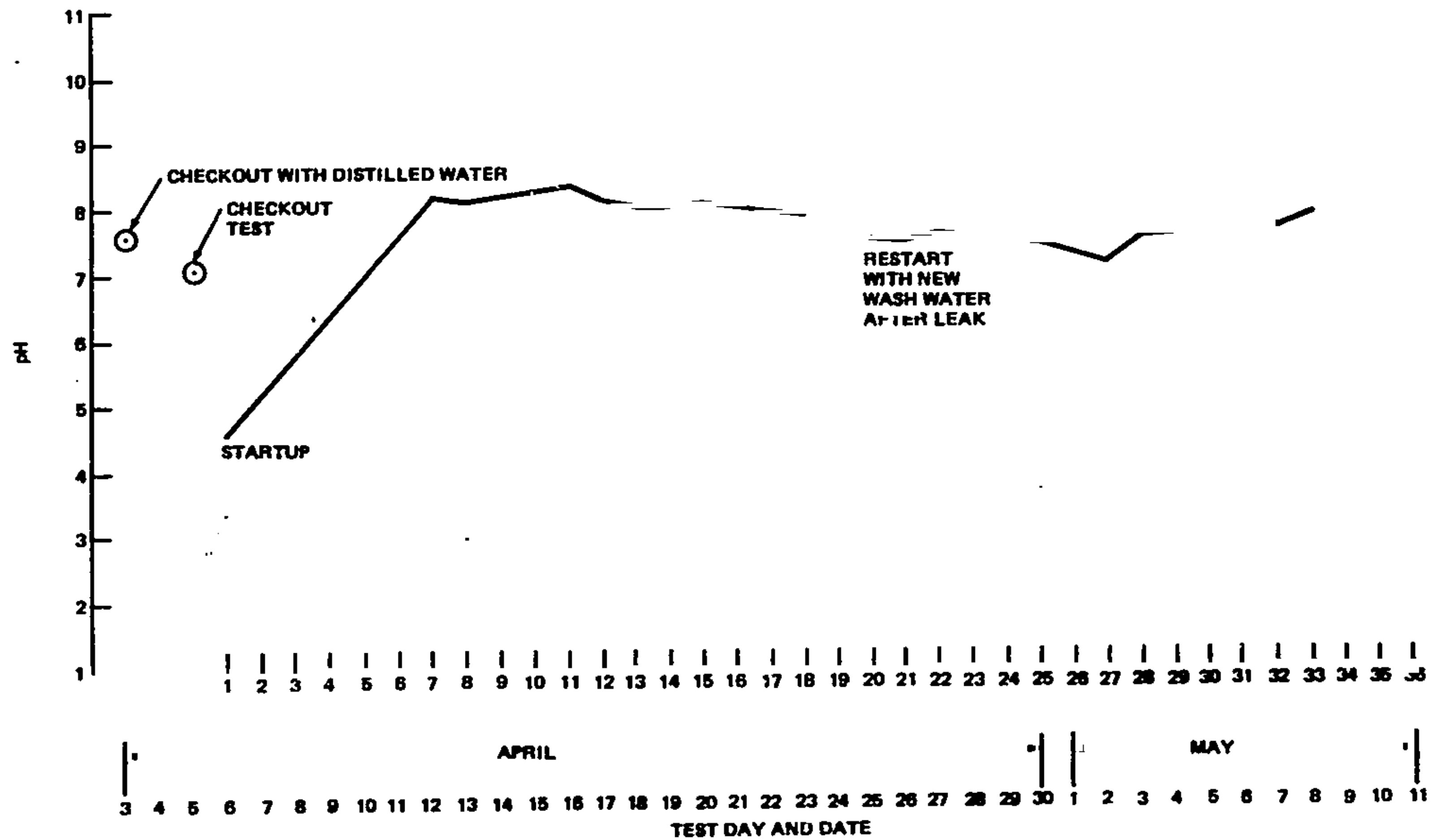


Figure 4-4. RO Pump Test pH

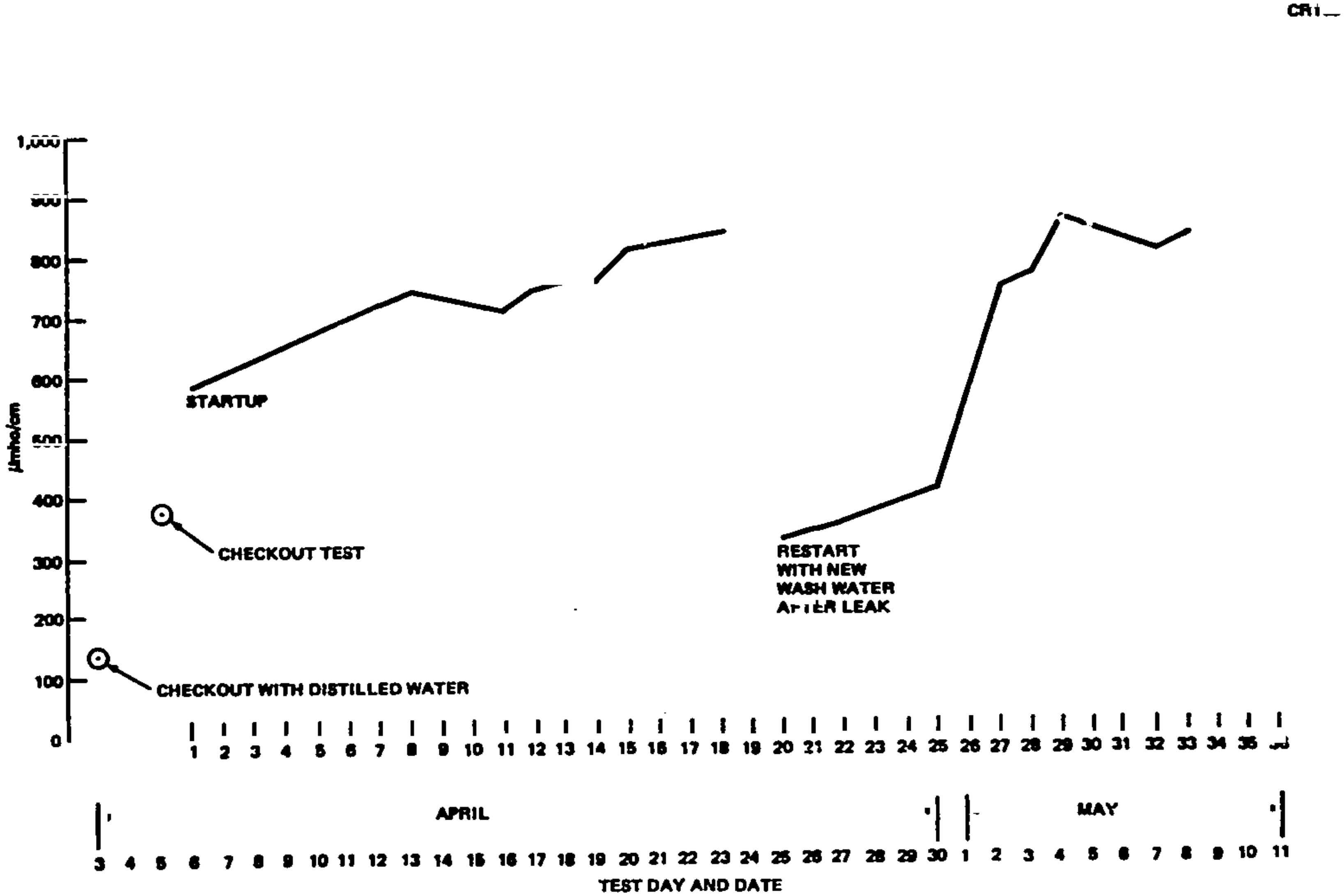


Figure 4-5. RO Pump Test Conductivity

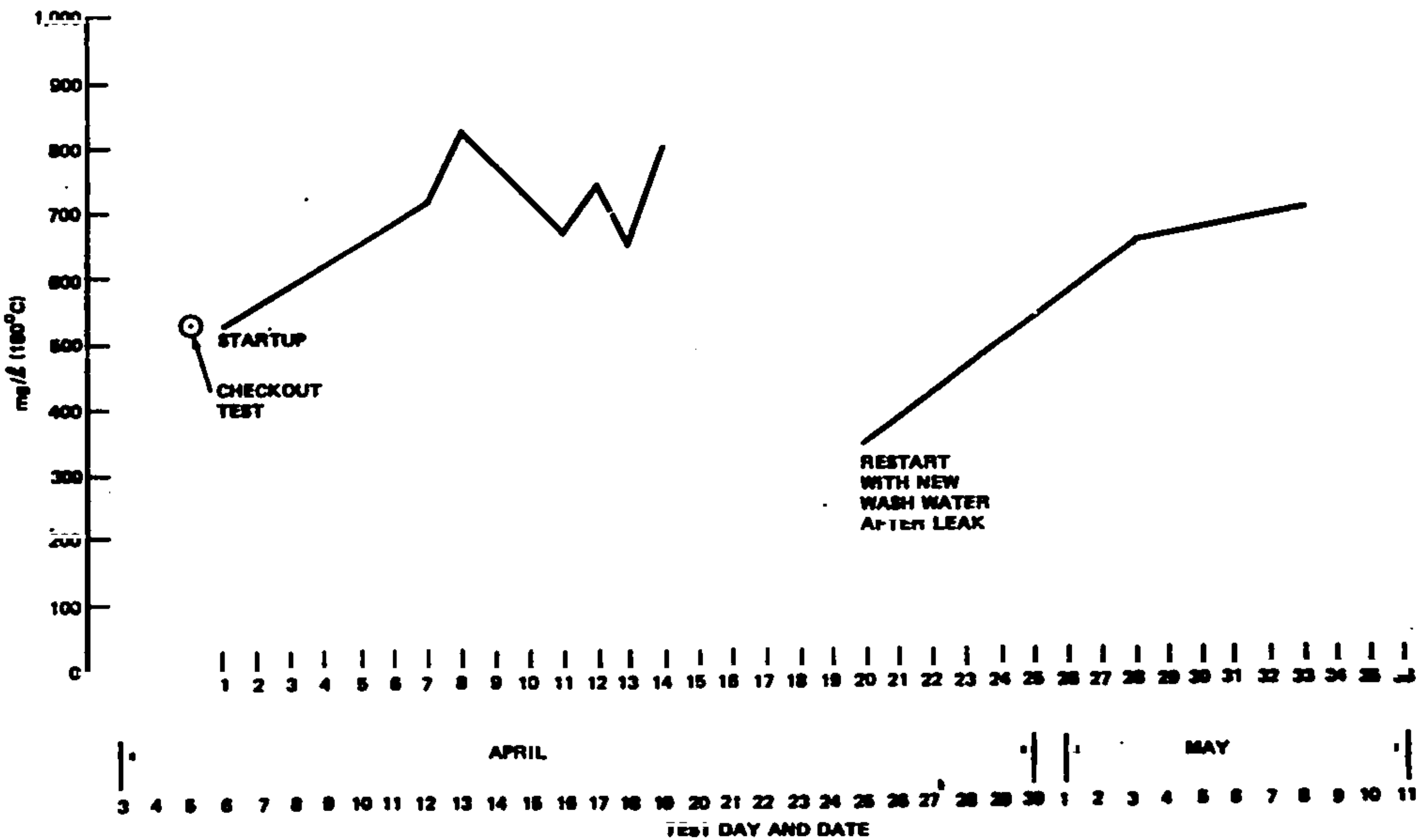


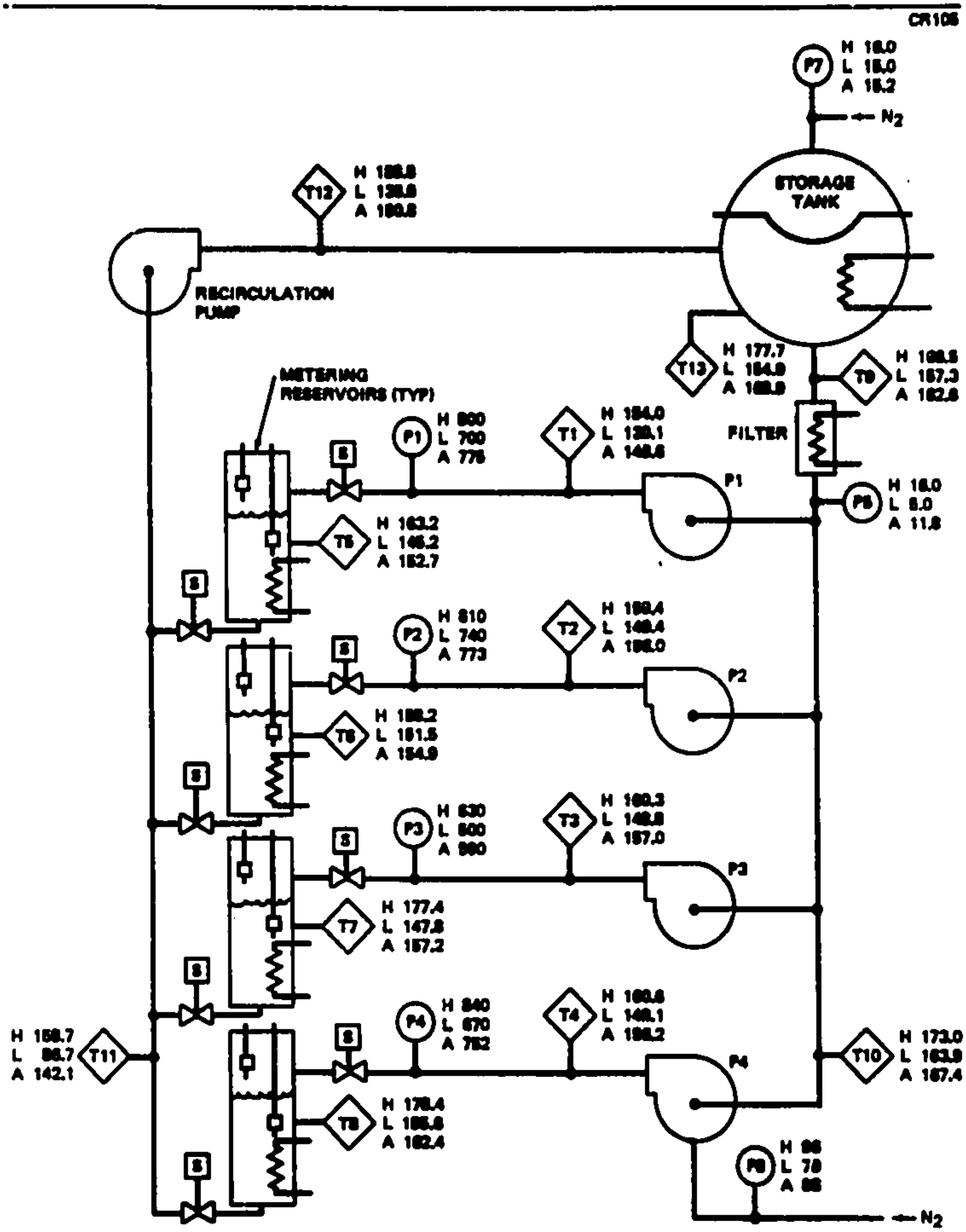
Figure 4-6. RO Pump Test TDS

Table 4-3

RO PUMP TEST CHEMICAL DATA

Analysis	Dates					
	4-5-73 Check-out	4-6-73 Start-up	4-18-73	4-25-73	5-3-73	5-8-73
TOC (mg/l)	320	200	480	240	340	390
Specific Conductivity (μmho/cm)	375	580	760	325	770	830
pH	7.1	4.7	8.0	7.5	7.5	7.8
Ammonia (mg/l)	0.29	0.67	0.44	0.37	0.30	0.37
Turbidity (ppm SiO_2)	142	35	9	20	4	4
Color (Pt-Co Units)	Colloid	Colloid	Blue* Hue	Blue* Hue	Blue* Hue	Blue* Hue
Foaming (in.)	2-1/2	3	3-1/2	2-1/2	4-1/2	5
Odor	Musty Rubber	Musty Rubber	Musty Rubber	Musty Rubber	Musty Rubber	Musty Rubber
TDS (100°F) (mg/l)	775	775	1,265	635	1,215	1,275
TDS (180°C) (mg/l)	525	521	640	333	644	688
Urea (mg/l)	0.0	1.97	2.0	1.9	2.1	2.0
Lactic Acid (mg/l)	0.7	1.3	24.2	23.6	22.3	38.0
NaCl (mg/l)	200	380	200	200	200	200
Sodium (Na) (mg/l)	50	87.5	85.5	40.0	93.0	102.0
Potassium (K) (mg/l)	18.8	50.0	36.5	12.5	38.8	48.5
Calcium (Ca) (mg/l)	5.4	5.6	2.9	1.5	3.1	2.9
Iron (Fe) (mg/l)	0.50	0.10	0.4	<0.2	<0.2	<0.2
Magnesium (Mg) (mg/l)	0.62	0.56	0.88	0.54	0.81	0.85
Chromium (Cr) (mg/l)	~0.06	~0.02	<0.04	<0.02	<0.03	<0.03

*As compared to (Pt-Co) Standards



LEGEND:

◇ INDICATES TEMPERATURE MEASUREMENTS (°F)

○ INDICATES PRESSURE MEASUREMENTS (PSIG)

H INDICATES HIGHEST READING TAKEN

L INDICATES LOWEST READING TAKEN

A INDICATES AVERAGE OF ALL READINGS TAKEN

Figure 4-7. RO Pump Test Pressure and Temperature Data Summary

Table 4-4

SUMMARY OF RO PUMP TEST RESULTS

	P1 Milton Roy	P2 BIF	P3 PCP	P4 Haskel
Reservoir Cycles	2265	2299	2122	776
Pump Cycles	24,718	13,639	23,885	928,115
Pump-On Time (hr)	542.94	600.58	386.90	321.55
Power Consumed by Pump (kwh)	143.3	178.5	91.0	---
Average Input Power (kw)	0.2639	0.2972	0.2352	---
Average Input Power (hp)	0.3539	0.3985	0.3154	0.4296*
Total Water Pumped (lb)	18,278.55	19,472.53	17,633.82	6,448.56
(gal)	2,202.23	2,346.09	2,124.56	776.93
Average Time per Pump Cycle (minutes/cycle)	1.32	2.64	0.97	.01667
Total Operating Time (hr)	699.08	711.58	639.28	294.23
No flow due to pumpout of other reservoirs by recirculation pump (hr)	30.58	31.04	28.65	10.48
Total Adjusted Operating Time (hr)	668.50	680.54	610.63	283.75
Average System Flow Rate (lb/min)	0.456	0.477	0.481	0.378
Average System Flow Rate (gal/min)	.0549	.0575	.0580	.0456
Average Pump Operating Press. (psig)	775	773	590	752
Average Pump Flow Rate (lb/min)	0.5611	0.5404	0.7596	0.3342
Average Output Power (hp)	0.0304	0.0292	0.0311	0.01758
Overall Efficiency (percent)	8.59	7.33	9.86	4.09
Motor Efficiency (percent)	23.5	45.5	48.8	NA**
Speed Reducer Efficiency (percent)	85	71	65	NA
Pump Efficiency (percent)	43.0	22.7	31.1	NA

*Assuming a 25 percent air compressor efficiency
**Not applicable

The average input power (Table 4-4) for the three electrically driven pumps was calculated from the individual elapsed time recorders and watt-hour meters as follows:

$$\text{Average Input Power (kw)} = \frac{\text{power consumed (kwh)}}{\text{pump-on time (hr)}}$$

The average input power for the air-driven pump was calculated from the total air usage, the inlet air pressure, and the pump duty cycle data recorded during the test. From these data it was determined that the average air usage was 0.6525 standard ft^3/min and the average pump inlet air pressure was 88 psig. The theoretical power requirements for the air-driven Haskel pump were calculated from the following relationship:

$$P' = \frac{k}{k-1}\left(\frac{144\ p_1\ v_1}{33 \times 10^3}\right)\left[\left(\frac{p_2}{p_1}\right)^{\frac{(k-1)}{k}} - 1\right]$$

where

P' = theoretical input power, hp

k = ratio of specific heats for air = 1.4

p_1 = ambient air pressure = 14.7 psia

p_2 = average pump inlet air pressure, psia

v_1 = average air usage, standard ft^3/min

Since:

p_2 = 88 + 14.7 = 102.7 psia

v_1 = 0.6425 ft^3/min

$$P' = \frac{1.4}{1.4-1}\left(\frac{144 \times 14.7 \times 0.6525}{33 \times 10^3}\right)\left[\left(\frac{102.7}{14.7}\right)^{\frac{(1.4-1)}{1.4}} - 1\right]$$

P' = 0.1074 hp

In order to make a valid comparison with the electrically driven pumps, the electrical horsepower required to drive an air compressor must be calculated. Assuming an air compressor/motor combined efficiency of 0.25 (Reference 11), the actual input power may be calculated:

$$\text{Actual Input Power} = \frac{0.1074}{0.25} = 0.4296 \text{ hp}$$

The toal system operating time for each pump was adjusted to compensate for the fact that a reservoir cycle of any pump would shut off flow at all pumps. Since an actual RO water recovery system would contain only one pump, the time required to pump out the other three pump reservoirs was subtracted from the total operating time of each pump. The total operating time was obtained from the test log and the time for reservoir cycles was obtained from the stripchart recorder. The average system flow rate was calculated from the total water pumped and the adjusted pump operating time.

Since each pump cycled on and off between the preset operating pressures, the pump duty cycle time was less than the total operating time and the average pump flow rate was greater than the system flow rate. This average pump flow rate and the average pump operating pressure were used to compute the average pump outlet power from the following relationship:

$$P_o = \frac{144\, p_o\, v_o}{33 \times 10^3}$$

where:

P_o = average output power, hp

p_o = average pump operating pressure, psig

v_o = average pump flow rate, lb/min

The overall pump efficiency may be calculated from the following relationship:

$$\text{Overall Efficiency (Percent)} = \frac{100\, P_o}{\text{Average Input Power (hp)}}$$

The motor efficiencies for the three electrically driven motors were obtained from data provided by the manufacturer which is shown in Figure 4-8. Additionally, the speed reduction efficiencies were provided by each pump

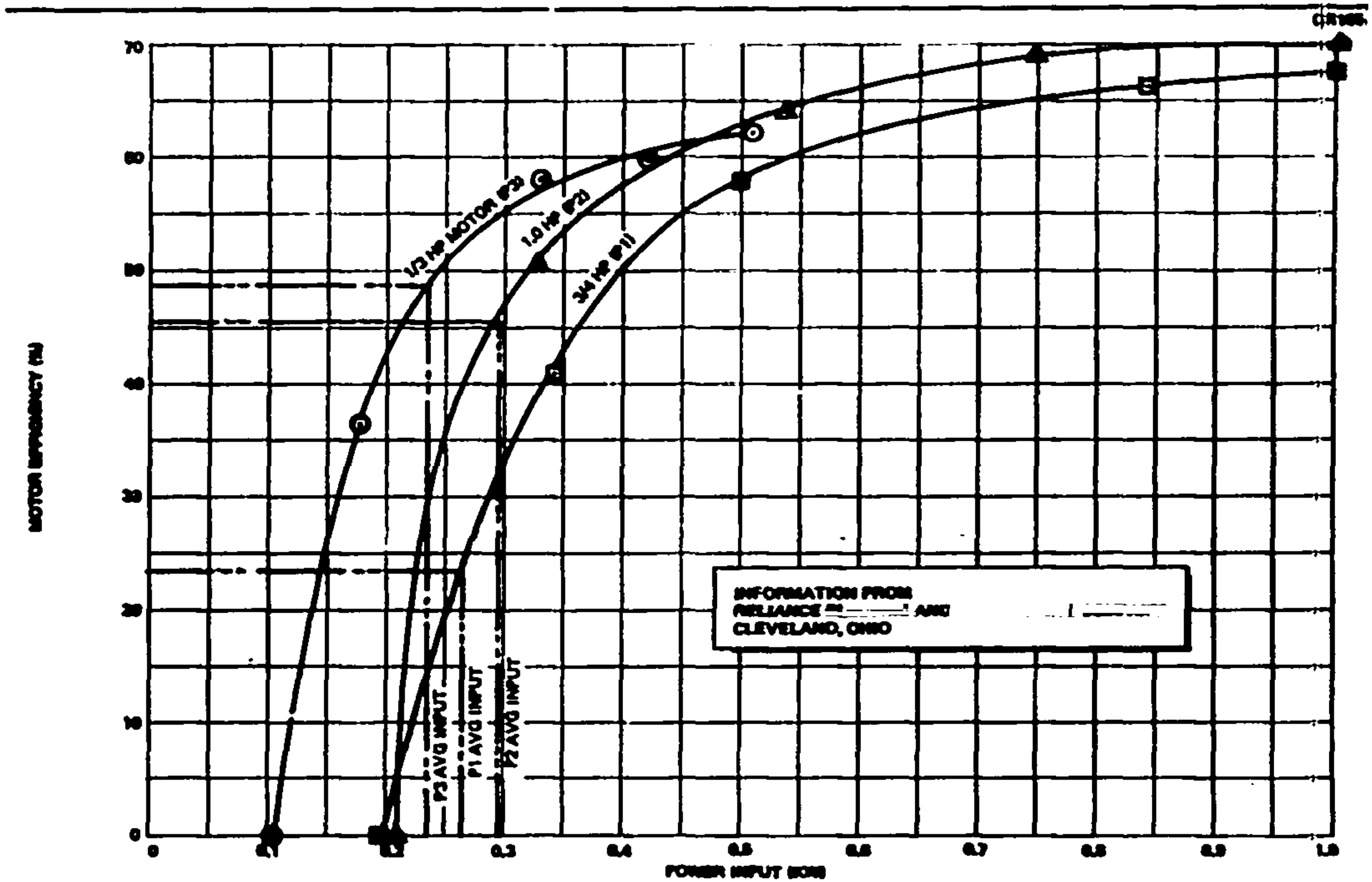


Figure 4-8. Pump Motor Performance

manufacturer and is the best information available. From these efficiencies the actual pump hydraulic efficiencies were calculated from the following relationships:

$$\eta_T = \eta_M \eta_G \eta_P ;$$

$$\eta_P = \frac{\eta_T}{\eta_M \eta_G}$$

where:

η_T = overall efficiency, percent
η_M = motor efficiency, percent
η_G = speed reducer efficiency, percent
η_P = pump efficiency, percent

An evaluation of each pump performance during the test and the results of the disassembly and inspection after the test, is discussed in the following paragraphs.

4.4.2.1 Test Performance and Posttest Evaluation of the Milton Roy Pump (P1)
The Milton Roy Pump, Model FR 141A-72, Serial No. 113509, is a crankshaft-driven, balanced-diaphragm pump. The pumping and control mechanism is illustrated in Figure 4-9. A summary of the pump operation during the 30-day test is shown in Table 4-5. This pump operated for 737.514 hours without failure and was shutdown 31.250 hours due to facility malfunctions and minor maintenance work.

The daily average flow rate of this pump is shown in Figure 4-10. The average flow rate for the entire test was 0.0549 gpm which was very close to the desired flow rate of 0.056 gpm. The major deviations in flow shown on Figure 4-10 were caused by facility malfunctions. Of these malfunctions, the deterioration of the accumulator bladder had the most severe effect on flow rate. This problem caused Visco Jet flow orifice clogging which necessitated constant metering valve adjustments and three replacements of the Visco Jet.

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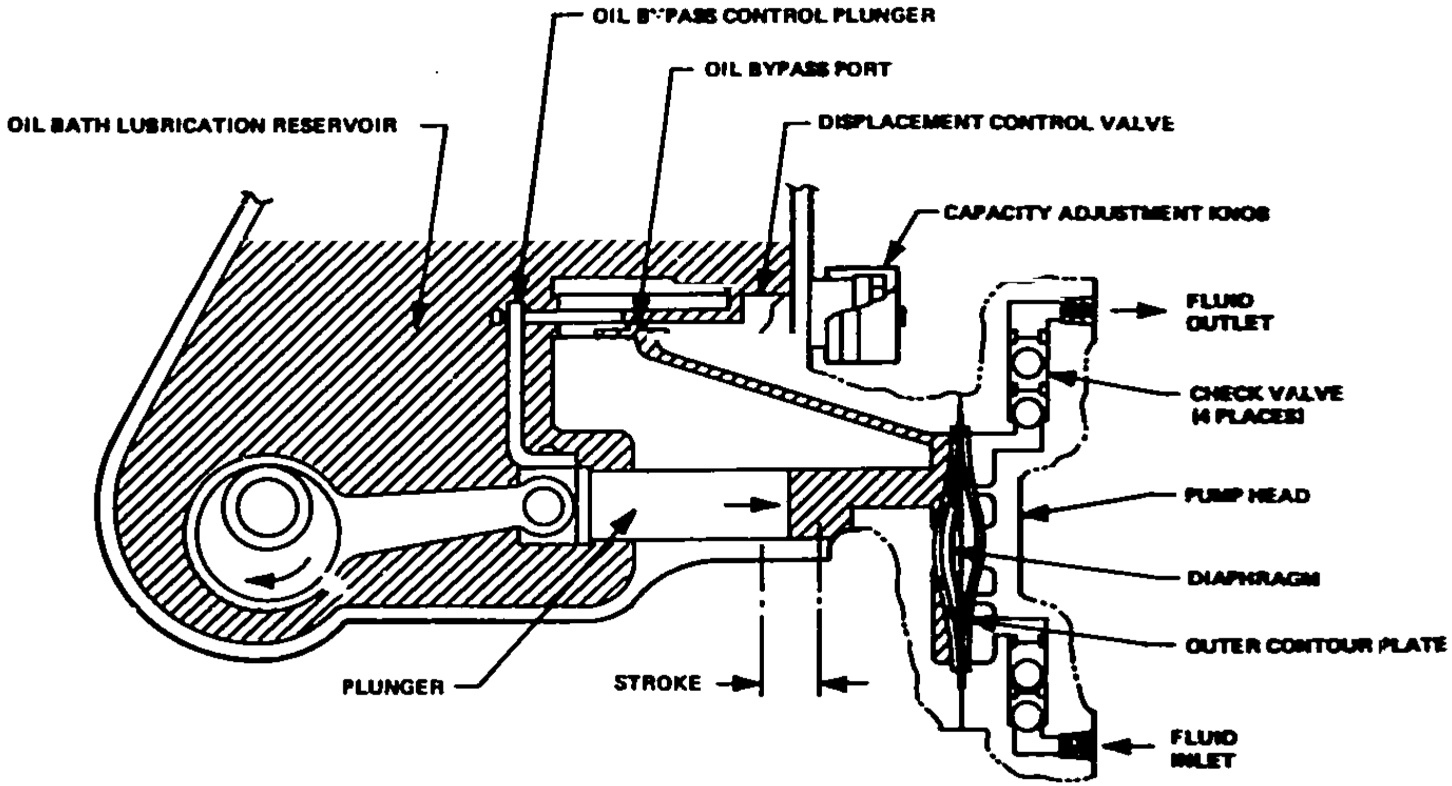

Figure 4-9. Milton Roy Model FR141A-72 Crankshaft-Driven Balanced Diaphragm Pump

Table 4-5

RO PUMP TEST – SUMMARY OF MILTON ROY PUMP OPERATION

Date	Start Time	Stop Time	Operating Time (hr)	Maintenance Time (hr)	Failure Time (hr)	Remarks
4-6-73	1534					
4-8-73		0933	41.983			
	1008			0.583		Replace fuse
4-9-73		1300	26.866			
	1315			0.250		Pressure switch installation for P4
4-10-73		0400	14.750			
	0800			4.000		Recirculation pump hangup
		1430	6.500			
	1510			0.667		Replace Visco Jet and solenoid valve
4-11-73		1520	24.167			
	1545			0.417		Replace system inlet filter
4-16-73		1505	119.333			
	1515			0.167		Install snubber on pressure switch
4-17-73		1124	20.150			
	1134			0.167		Change snubber
4-18-73		1009	22.583			
	1015			0.100		Change Visco Jet
4-24-73		0756	141.683			
	1637			8.683		Pressure switch leak
4-25-73		0855	16.300			
	1218			3.383		Add water
4-26-73		2100	32.700			
4-27-73	0750			10.833		Overpressure shutdown
		1141	3.850			
	1300			1.317		Replace accumulator, Visco Jet, and metering valve

Table 4-5

RO PUMP TEST – SUMMARY OF MILTON ROY PUMP OPERATION (Continued)

Date	Start Time	Stop Time	Operating Time (hr)	Maintenance Time (hr)	Failure Time (hr)	Remarks
4-27-73		1326	0.433			
	1352			0.433		Replace pressure switch
5-2-73		1500	121.133			
	1515			0.250		Replace accumulator and Visco Jet
5-8-73		1620	145.083			Test complete
			737.514	31.250	0	Totals
			31.250			
			768.764			Total Test Duration

The operation of this pump was generally smooth. However, the pump did exhibit outlet pressure fluctuations which were slightly higher than the other pumps. Because of these fluctuations, a pressure snubber was installed on the high-pressure control switch line. The pressure switch failure on day 19 was attributed to pressure fluctuations which fatigued the pressure switch bellows.

After completion of the 30-day test, the pump inlet and outlet valve assemblies were unscrewed and inspected. They were found to be clean and free from wear. The pump head was removed and the working chamber was found to contain a few fibers and considerable residue. Figure 4-11 shows the residue in the fluid passage. The passage walls were noted to be quite rough resulting from a light, incomplete grinding of the casting surface. This condition provided an excellent surface for the collection of debris. These unground wall areas collected most of the residual material.

The water side of the outer contour plate (see Figure 4-9) also collected a similar deposit to that shown in Figure 4-12. The diaphragm is shown in Figure 4-13 for the water side and in Figure 4-14 for the oil side. The drive end parts, motor coupling, and motor were in good condition with no visible wear, distortion, or broken parts.

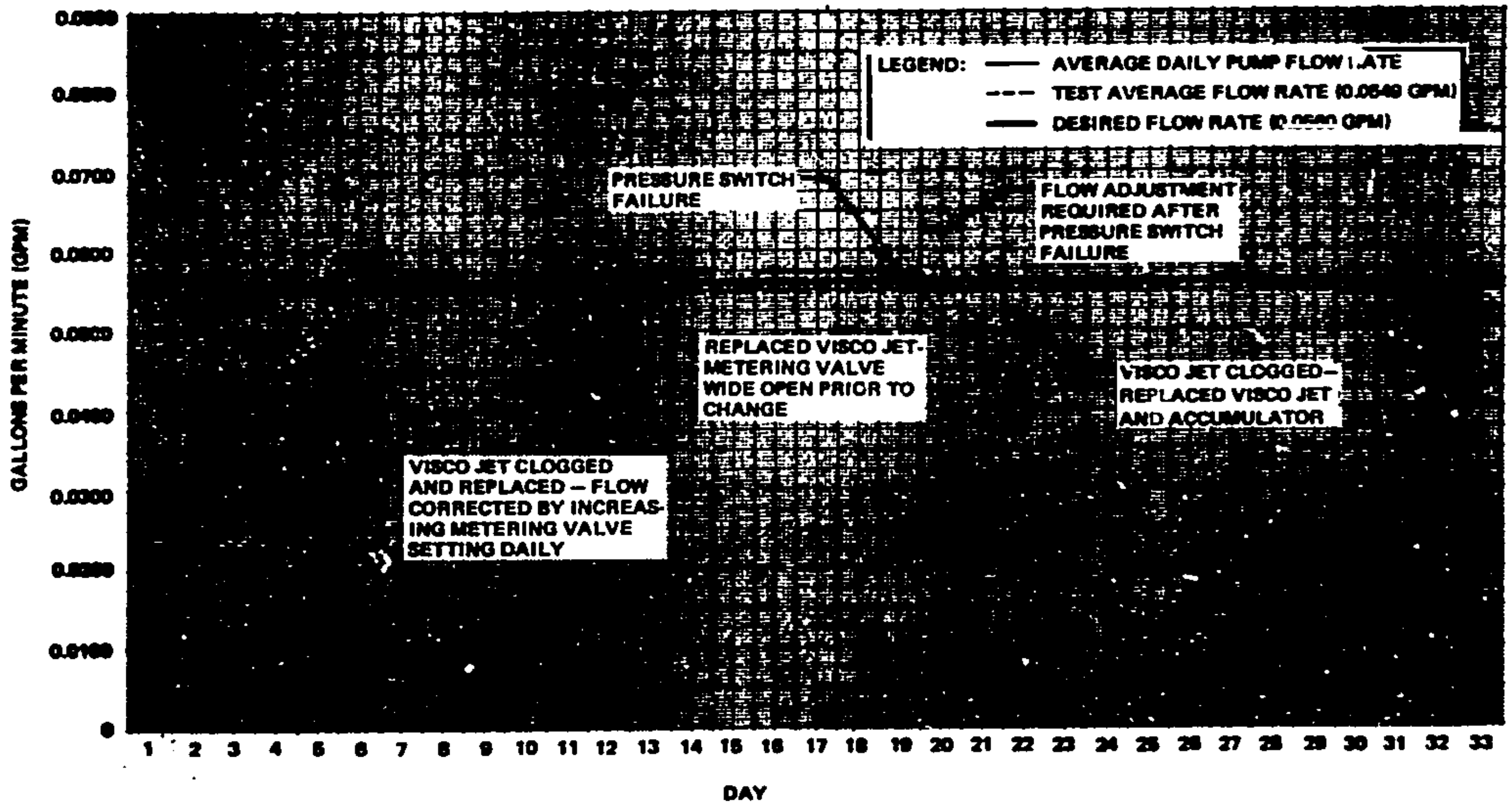

Figure 4-10. Daily Average Milton Roy Pump Flow Rates

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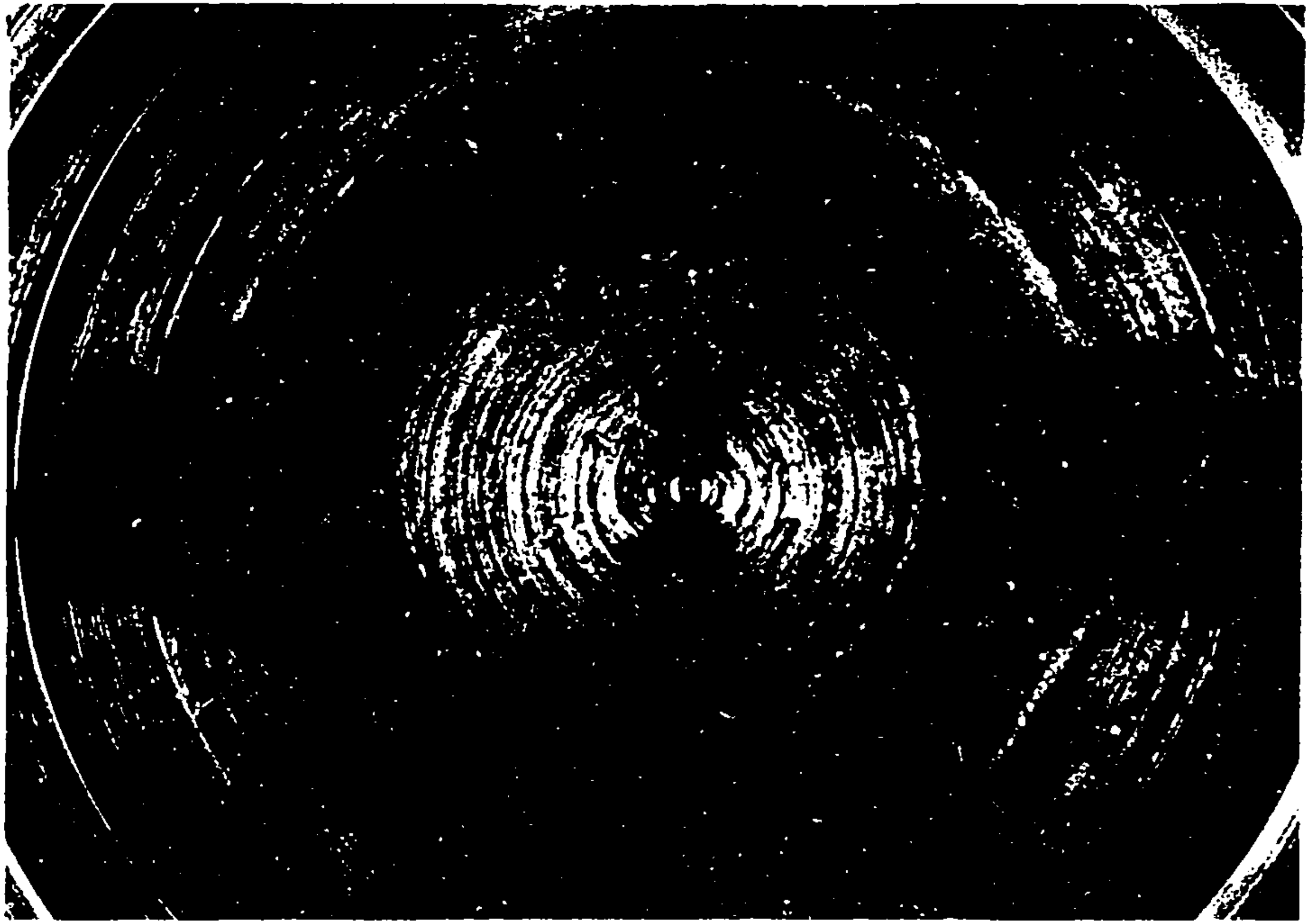

Figure 4-11. Inside View of Milton Roy Pump (P1) Head After Test

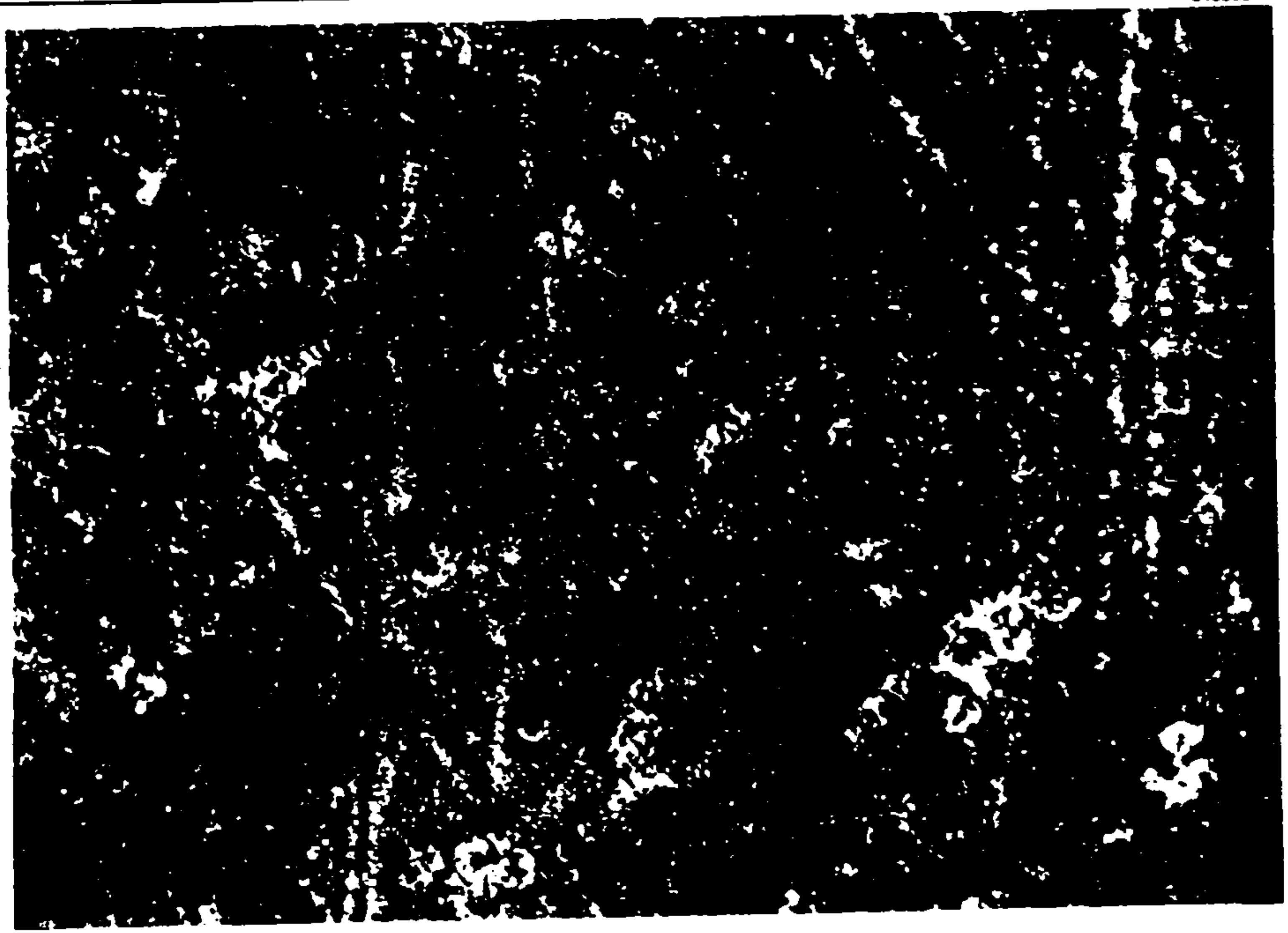

Figure 4-12. Water Side of Milton Roy Pump (P1) Outer Contour Plate After Test

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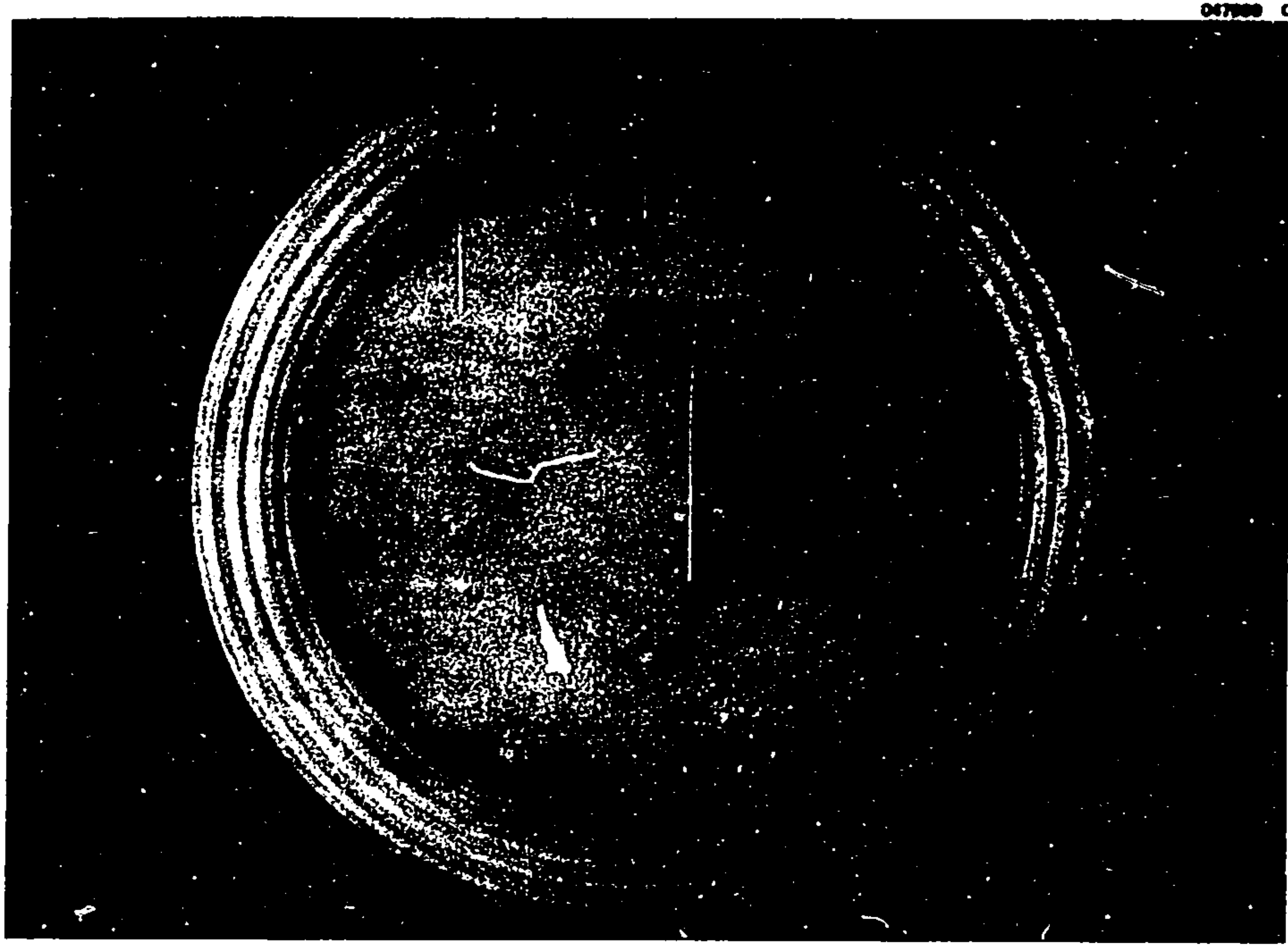

Figure 4-13. Water Side of Milton Roy Pump (P1) Diaphragm After Test

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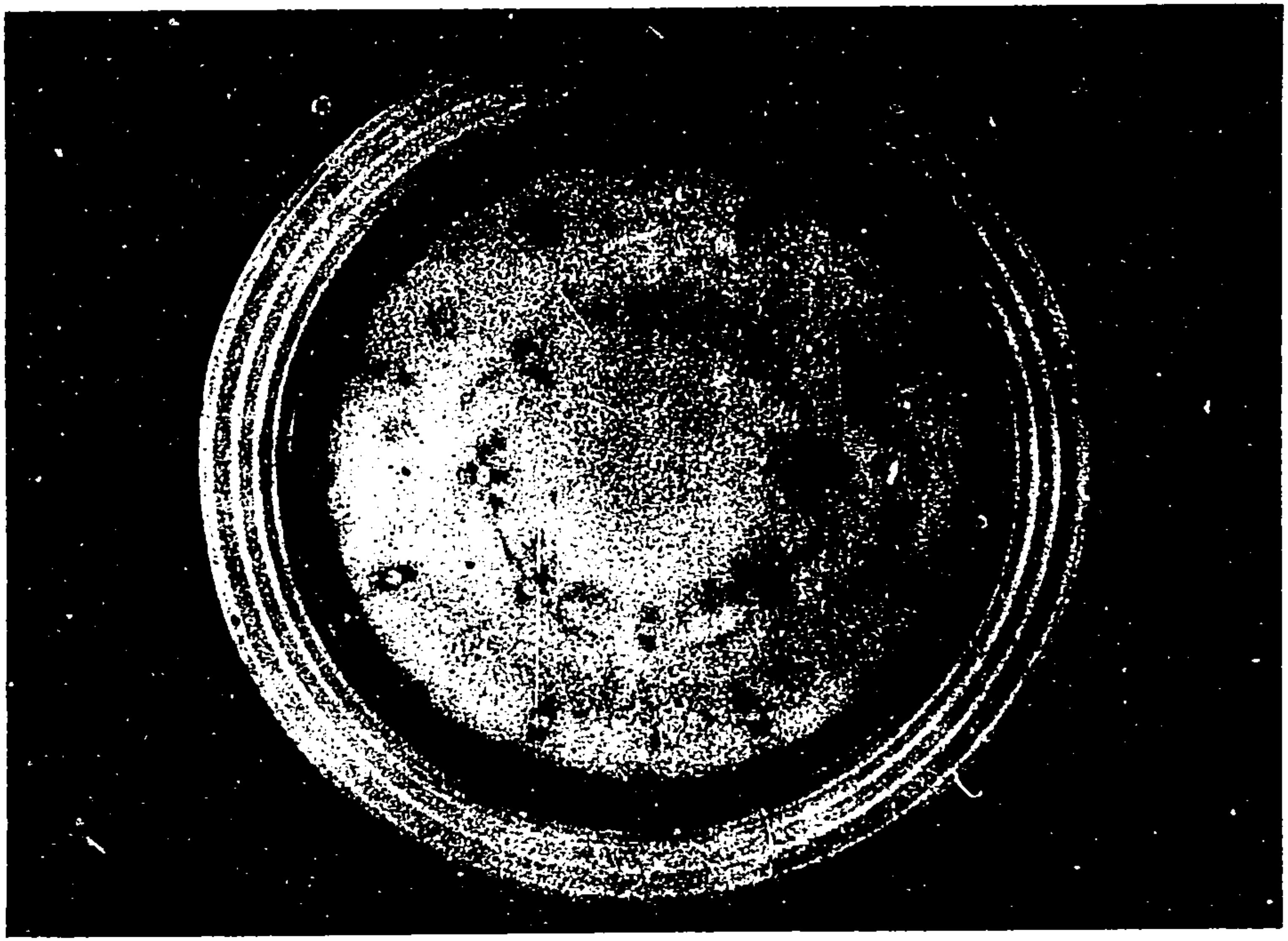

Figure 4-14. Oil Side of Milton Roy Pump (P1) Diaphragm After Test

4.4.2.2 Test Performance and Posttest Evaluation of the BIF Pump (P2)

The BIF pump, Model 1731-12-9511, Serial No. 21216-1, is an crankshaft-driven balanced tubular diaphragm pump. The pumping and control mechanism is illustrated in Figure 4-15. A summary of the pump operation during the test is shown in Table 4-6. This pump operated for 750.332 hours without failure and was shut down 18.434 hours due to facility malfunctions and maintenance only.

The daily average flow rate of this pump is shown in Figure 4-16. The average flow rate for the entire test was 0.0575 gpm which slightly exceeded the desired flow rate of 0.056 gpm. The major deviations in flow shown in Figure 4-16 were caused by facility malfunctions. The deterioration of the accumulator had less effect on this pump than on either P1 or P3. This problem did cause some flow orifice clogging, but the Visco Jet was only changed twice during the test. The accumulator bladder was changed on day 26, but this was only a precautionary measure since the bladder had not completely failed.

The operation of this pump was the most trouble free, and the pump output pressure pulsating the least of all four pumps tested. This smoothness of operation probably contributed to the long pump-on cycle of 2.64 minutes which was achieved with this pump (see Table 4-4) and may also have contributed to better accumulator bladder life.

After completion of the test, an initial external inspection of this pump revealed that the gear box shaft seal was leaking. Disassembly of the liquid-end check valves followed the external examination. There was some debris and deposits throughout the valve units. There were small amounts of metal chips on the concave sections of the ball guides and fiber particles and soap residues on the seats, guides, and walls. There was no indication of wear. There were considerably more deposits and particles in this pump's valve assemblies than in any other test pump.

The O-ring seals in the check valve bodies (Figure 4-15) all had pinched edges. The frayed edges of the O-rings can be seen in Figure 4-17. The grooves are

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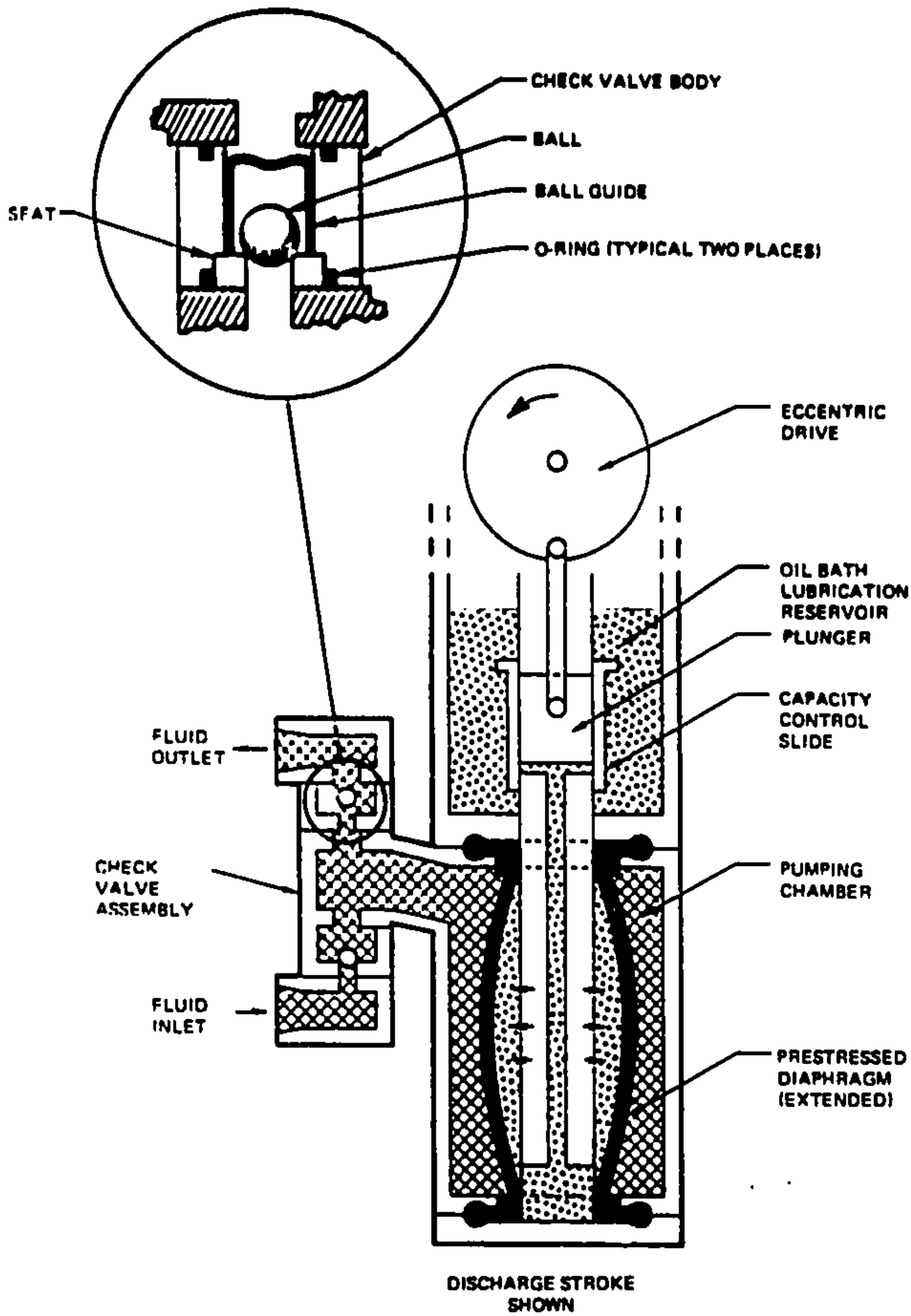

Figure 4-15. BIF Model 1731-12-9511 Crankshaft-Driven Balanced Tubular Diaphragm Pump

Table 4-6

RO PUMP TEST – SUMMARY OF BIF PUMP OPERATION

Date	Start Time	Stop Time	Operating Time (hr)	Maintenance Time (hr)	Failure Time (hr)	Remarks
4-6-73	1534					
4-9-73		1300	69.433			
	1315			0.250		Pressure switch installed for P4
4-10-73		0400	14.750			
	0800			4.000		Recirculation pump hangup
		1430	6.500			
	1510			0.667		Replace No. 1 Visco Jet and Solenoid valve
4-11-73		1520	24.167			
	1545			0.417		Replace system inlet filter
4-13-73		1503	47.300			
	1512			0.150		Change Visco Jet
4-18-73		1520	120.133			
	1600			0.667		Oil change at 240 hours
4-24-73		0756	135.933			
	1637			8.683		Pressure switch leak in P1
4-25-73		0855	16.300			
	1218			3.383		Add water
5-1-73		1624	148.100			
	1637			0.217		Replaced accumulator and Visco Jet
5-8-73		1620	167.716			Test completed
			750.332	18.434	0	Totals
			18.434			
			768.766			Total Test Duration

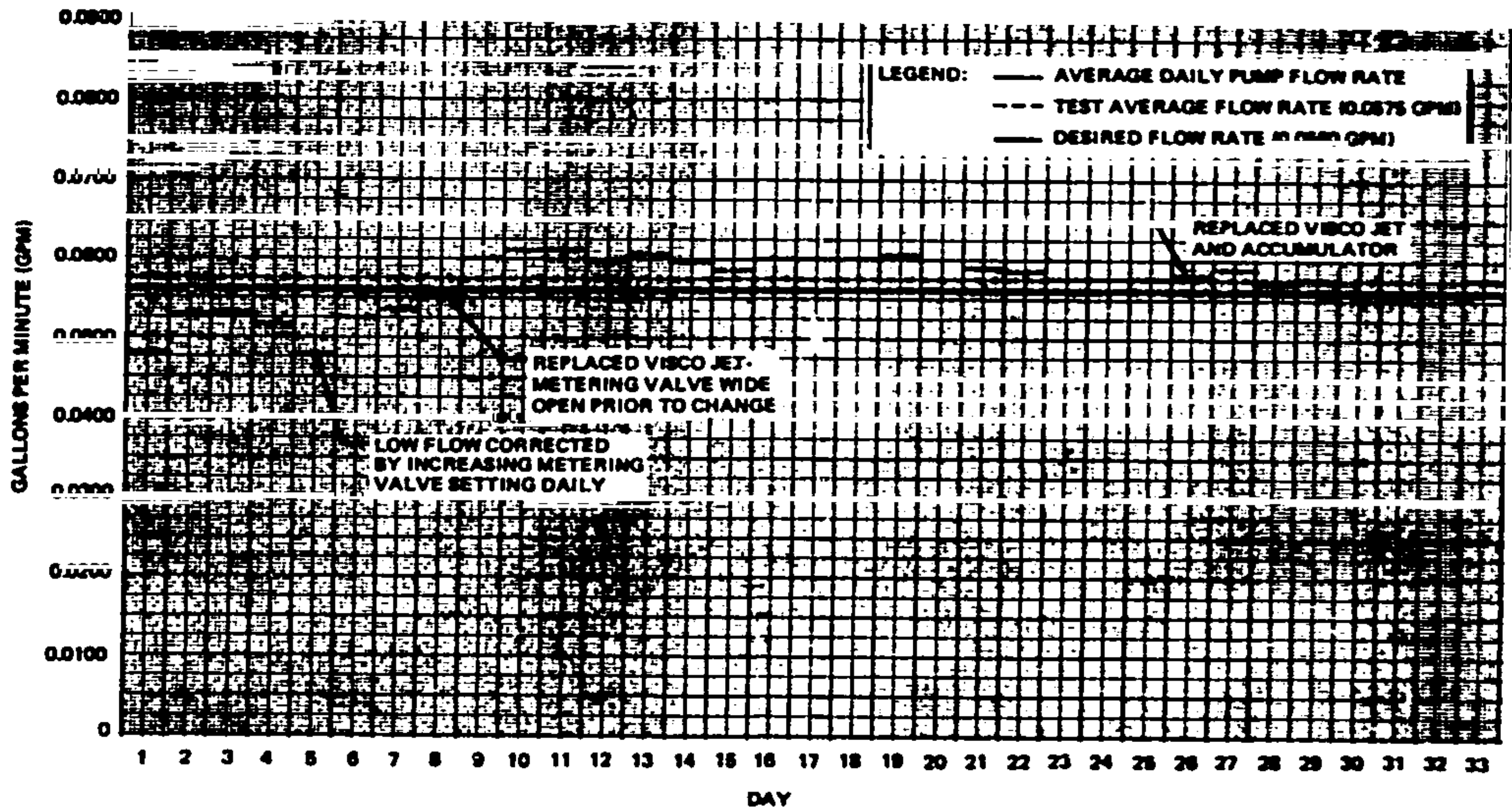

Figure 4-16. Daily Average BIF Pump Flow Rates

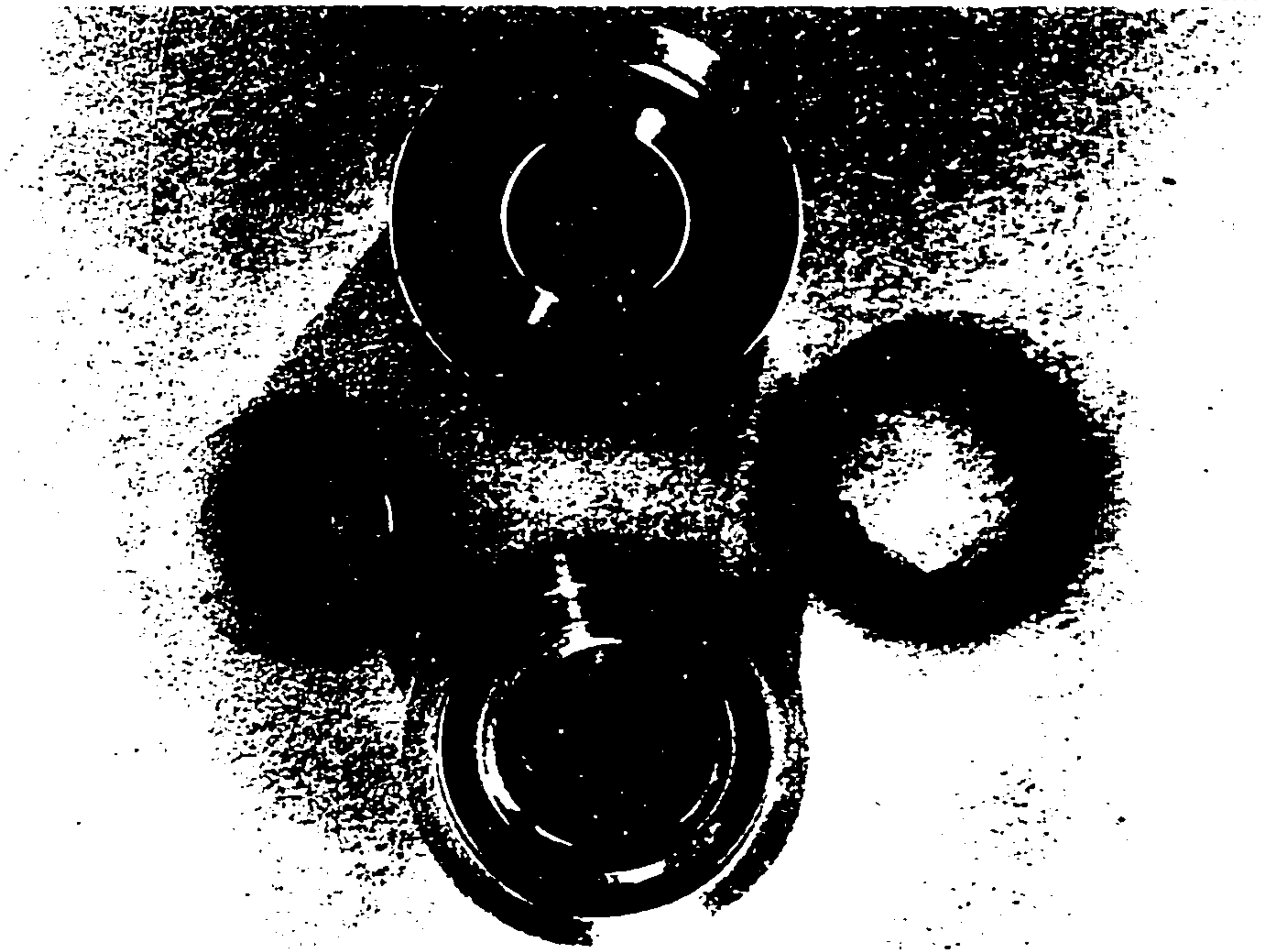

Figure 4-17. Check Valve Body O-Ring Seals from BIF Pump (P2)

sized so that a considerable amount of the O-ring projects above the groove. When the units are assembled and tightened, the O-rings are pinched between the mating surfaces.

The measuring cylinder assembly was disassembled and inspected for wear and debris. There were no particles or residues in any of the passages and no discernible wear. The diaphragm seemed in excellent condition although the lower edge was slightly frayed. This very probably occurred at assembly.

4.4.2.3 Test Performance and Posttest Evaluation of the PCP Pump (P3)
The Precision Control Products Pump, Model 1971-121, Serial No. 72121113, is an eccentric-driven unbalanced diaphragm pump. The pumping and control mechanism is illustrated in Figure 4-18. The pump incorporates an advanced waveform, or harmonic, drive which is shown in Figure 4-19. The motor directly drives a ball bearing with an inner race that is eccentric and an outer race that is flexible. This bearing slightly flexes a many-toothed spline inside a mating ring which has four more teeth. As a result of the different number of teeth, the flex spline slowly rotates and turns the stroking eccentric.

A summary of the pump operation during the test is shown in Table 4-7. This pump operated for 675.932 hours before failure. Prior to final failure, this pump was shut down for 24.049 hours due to pump malfunctions and for 68.785 hours due to facility maintenance and malfunctions.

The daily average flow rate of this pump is shown in Figure 4-20. The average flow rate for the entire test was 0.058 gpm which slightly exceeded the desired flow rate of 0.056 gpm. This high flow rate was achieved in spite of the many pump malfunctions and the facility malfunctions which occurred with this pump. There were three failures of the pump outlet check valve O-ring. After each of the first two failures on days 2 and 5, the O-ring was replaced with the PCP recommend spare, Part No. 996. After the third failure on day 7, an engineering evaluation was made after discussions with PCP personnel. It was decided to replace the PCP O-ring with a MS28778-8 O-ring which was similar in outer diameter but has a smaller cross-sectional area. This modification eliminated the O-ring problem.

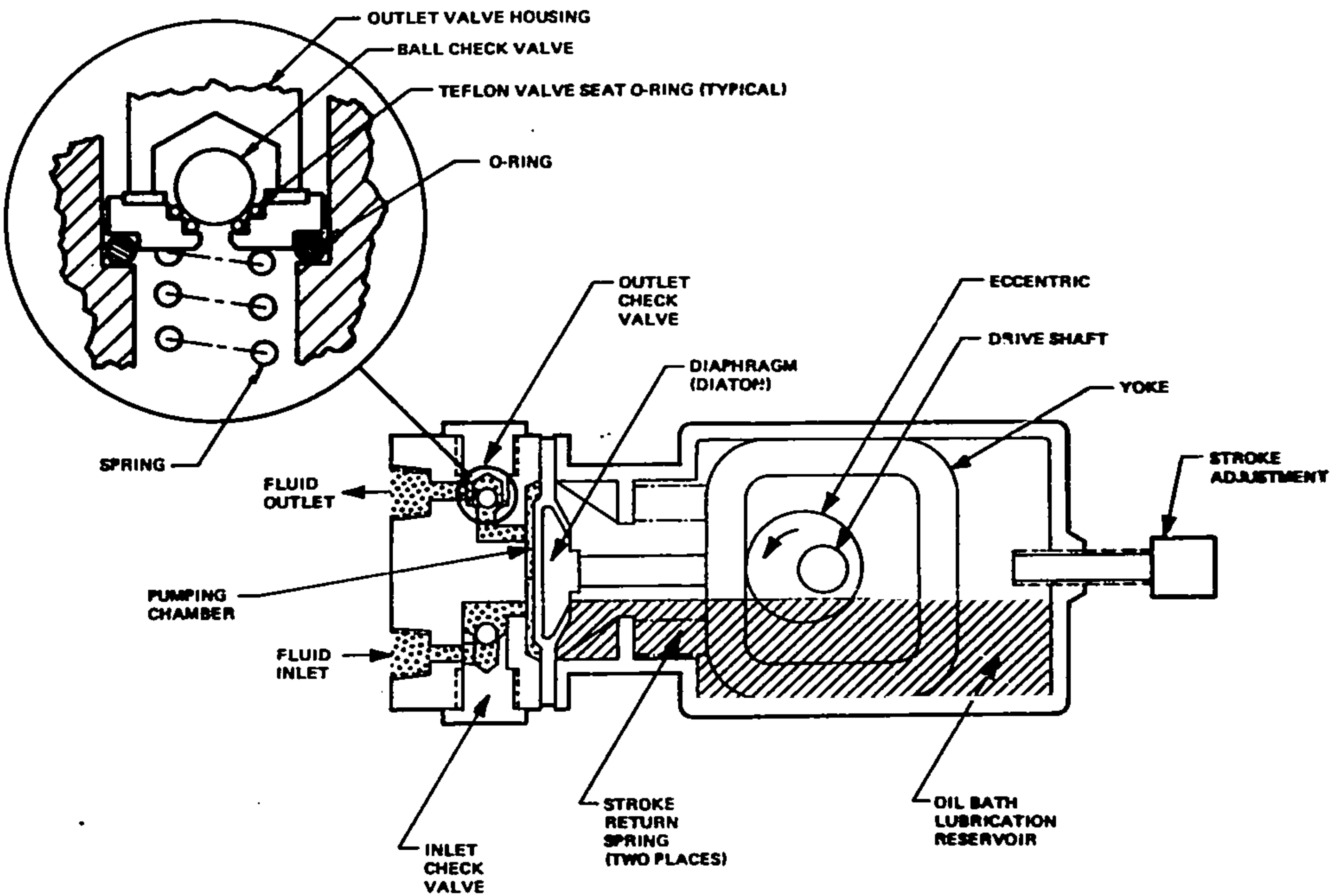

Figure 4-18. PCP Model 1971-121 Eccentric-Driven Unbalanced Diaphragm Pump

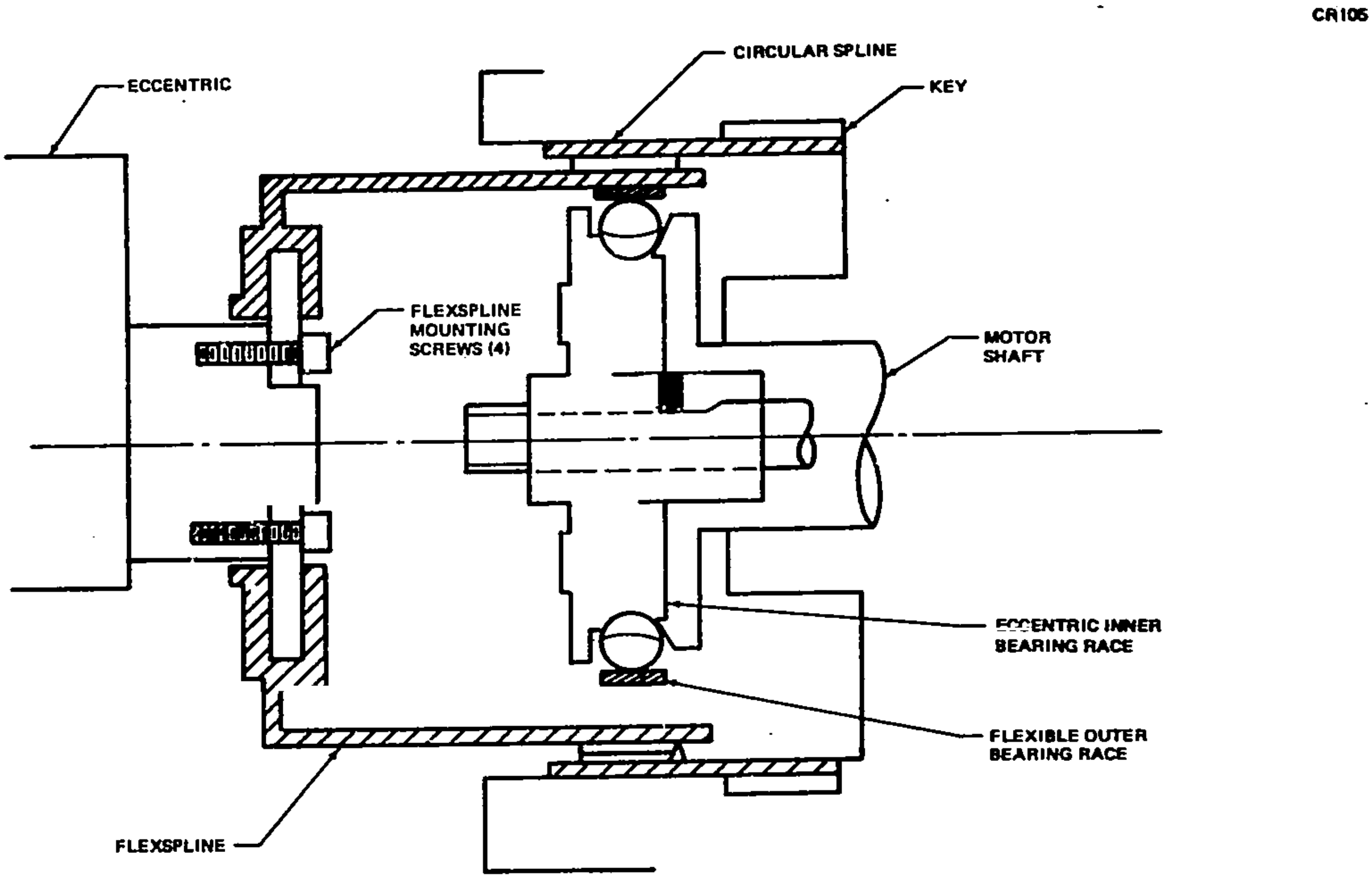

Figure 4-19. PCP Pump Waveform (Harmonic) Drive

Table 4-7

RO PUMP TEST – SUMMARY OF PCP PUMP OPERATION

Date	Start Time	Stop Time	Operating Time (hr)	Maintenance Time (hr)	Failure Time (hr)	Remarks
4-6-73	1534					
		2039	5.083			
4-7-73	1313				16.566	O-ring failed
4-9-73		1300	47.783			
	1315			0.250		Pressure switch installation for P4
4-10-73		0400	14.750			
	0855			4.000	0.917	Recirculation pump hangup; O-ring failed
		1430	5.583			
	1510			0.667		Replace No. 1 Visco Jet and solenoid valve
4-11-73		1520	24.167			
	1545			0.417		Replace system inlet filter
4-12-73		0823	16.633			
	1428				6.083	O-ring failed
4-24-73		0745	281.466			
4-25-73	1218			28.367		Pressure switch leak in P1; accumulator failure; add water
5-5-73		2126	249.133			
5-7-73	0806			34.667		Overpressure shutdown
5-8-73		1018	26.200			
	1035			0.284		Overpressure shutdown
		1130	0.917			
	1138				0.133	Problem troubleshooting change O-ring
		1155	0.284			
	1208				0.217	Problem troubleshooting replace check valve parts

Table 4-7

RO PUMP TEST – SUMMARY OF PCP PUMP OPERATION (Continued)

Date	Start Time	Stop Time	Operating Time (hr)	Maintenance Time (hr)	Failure Time (hr)	Remarks
5-8-73		1220	0.200			
	1228				0.133	Problem troubleshooting replace grease
		1516	2.800			
	1524			0.133		Calibrate pressure transducer
		1620	0.933			Final pump failure
			675.932	68.785	24.049	Totals
			68.785			
			24.049			
			768.766			Total Test Duration

After the O-ring problem was corrected, the pump operated normally for 281.466 hours until the test stand was shut down due to the pressure switch leak on day 19. When the test stand was restarted, it was discovered that the No. 3 accumulator bladder had failed. This was the first accumulator failure and was due to the use of a bladder constructed of polyurethane rubber as previously discussed. During the investigation of this malfunction, the O-ring, which had been installed on day 7, was replaced as a preventive maintenance measure. The O-ring had not failed but was cut slightly on the inner circumference.

After replacement of the accumulator and Visco Jet, the pump operated normally for 249.133 hours until the No. 3 system automatic overpressure shut down on day 30. Since this shut down occurred on a weekend, it was not detected until day 32. It was suspected that the overpressure condition was caused by a drifting pressure switch setting, which was corrected.

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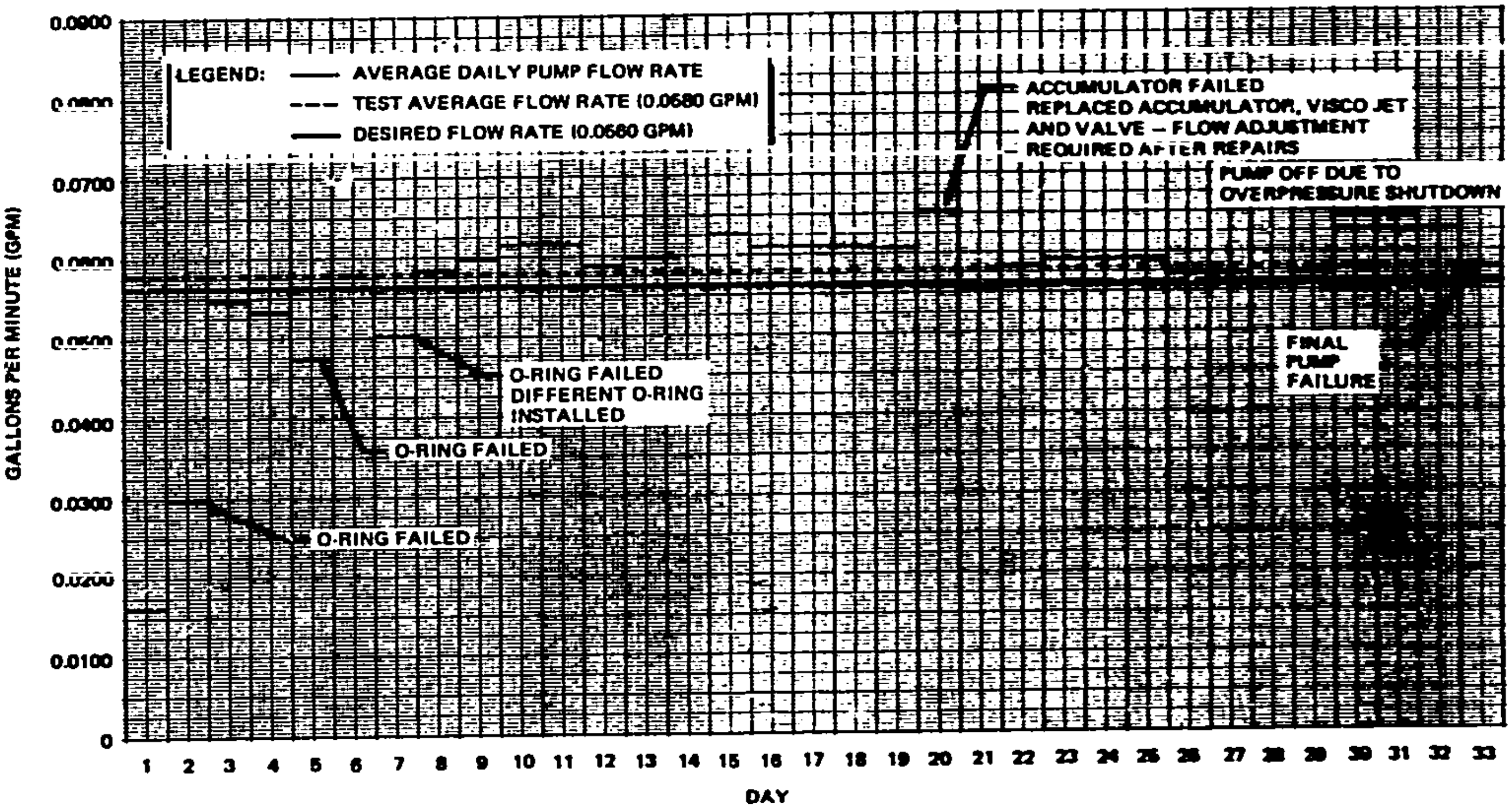

Figure 4-20. Daily Average PCP Pump Flow Rates

The pump developed a slight but noticeable clicking noise which was first detected at 0815 on day 33. Additionally, the pump operation became erratic and overpressure shutdown occurred with an outlet pressure of 600 psig. (The overpressure shut down switch had been set to actuate at 700 psig.) Attempts to adjust the pressure switches did not improve the operation, which indicated that pressure surges may have been present in the pump discharge. The O-ring and both inlet and outlet check valve parts were replaced with no change in operation. The gear case cover was removed and it was discovered that the lithium grease had leaked out of the harmonic drive case into the oil crankcase. The grease was replaced and the mechanism inspected, but the cause of the clicking noise was not discovered and operation did not improve.

In order to evaluate the magnitude of the suspected P3 pump discharge pressure transients, the pressure transducers on the three operating pumps (P4 had failed on day 18) were calibrated and connected to a recording oscilloscope. Typical pressure transients are shown in Figure 4-21. As can be seen in Figure 4-21, P3 had developed a severe pressure transient having a lower peak at approximately 100 psig and an upper peak at approximately 1,200 psig. The pressure transients for P1 and P2 did not exhibit any severe transients. The P3 pump was considered failed and shut down for posttest disassembly and evaluation.

Disassembly of the head and valve assemblies was the first step of the pump inspection. There were no deposits or debris on the parts or in the passages. The MDAC-installed O-ring, MS28778-8, which had replaced the PCP Part No. 996 O-ring on day 7, had a cut at the inner side for 360 degrees of its circumference as shown in Figure 4-22. There was no evidence of mechanical wear on any of the metal surfaces.

The diaphragm was stuck to the head and both were unscrewed from the shaft by rotating the head. The rubber-to-metal bond at the diaphragm shaft had torn loose all around as shown in Figure 4-23.

The diaphragm was stuck to the head so that they could not be broken apart by lateral pressure while in a vise or from blows of a mallet. Inserting a knife blade into the joint all around the diaphragm finally separated the two

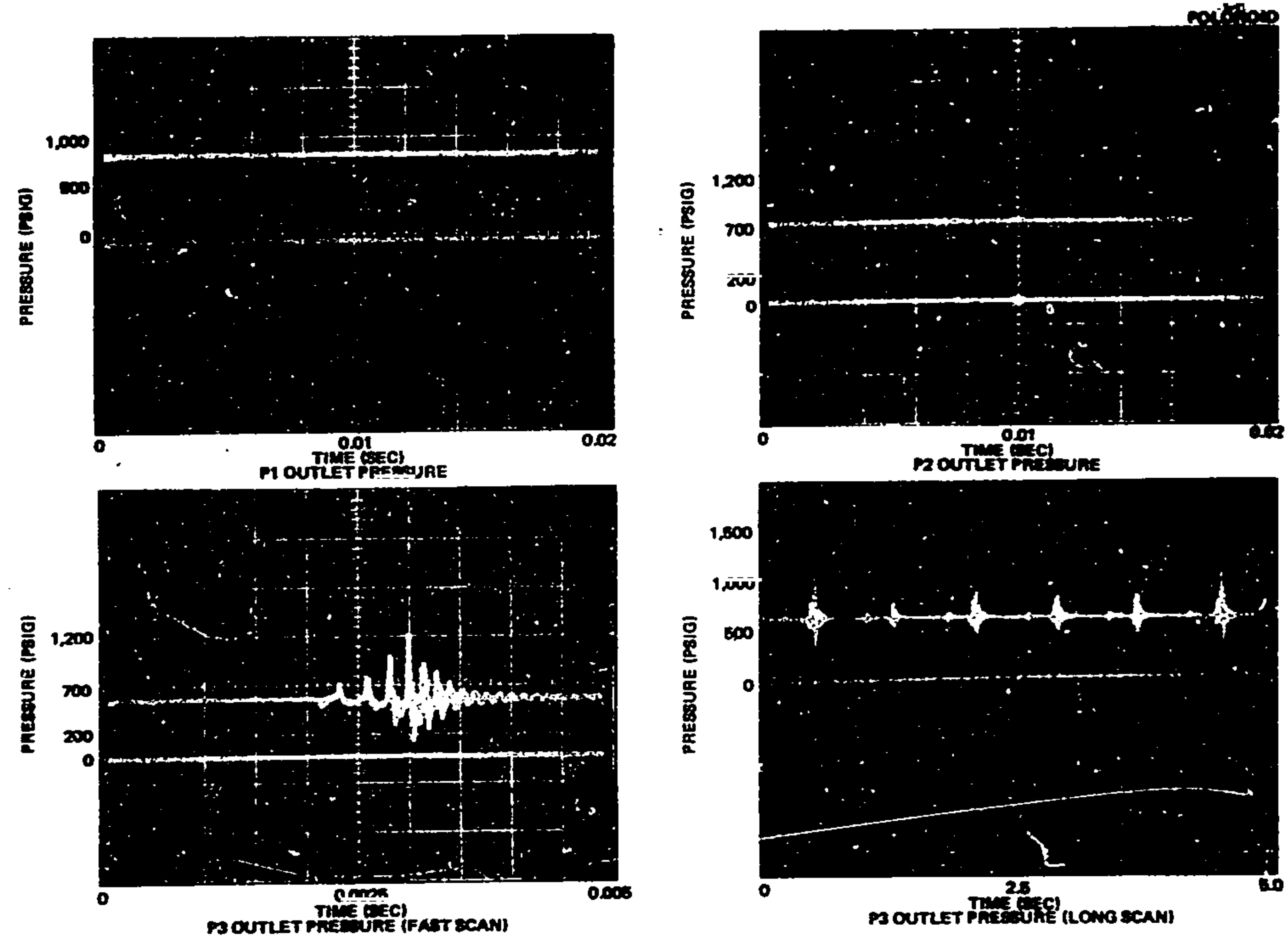

Figure 4-21. Comparison of P1, P2, and P3 Outlet Pressure Transients

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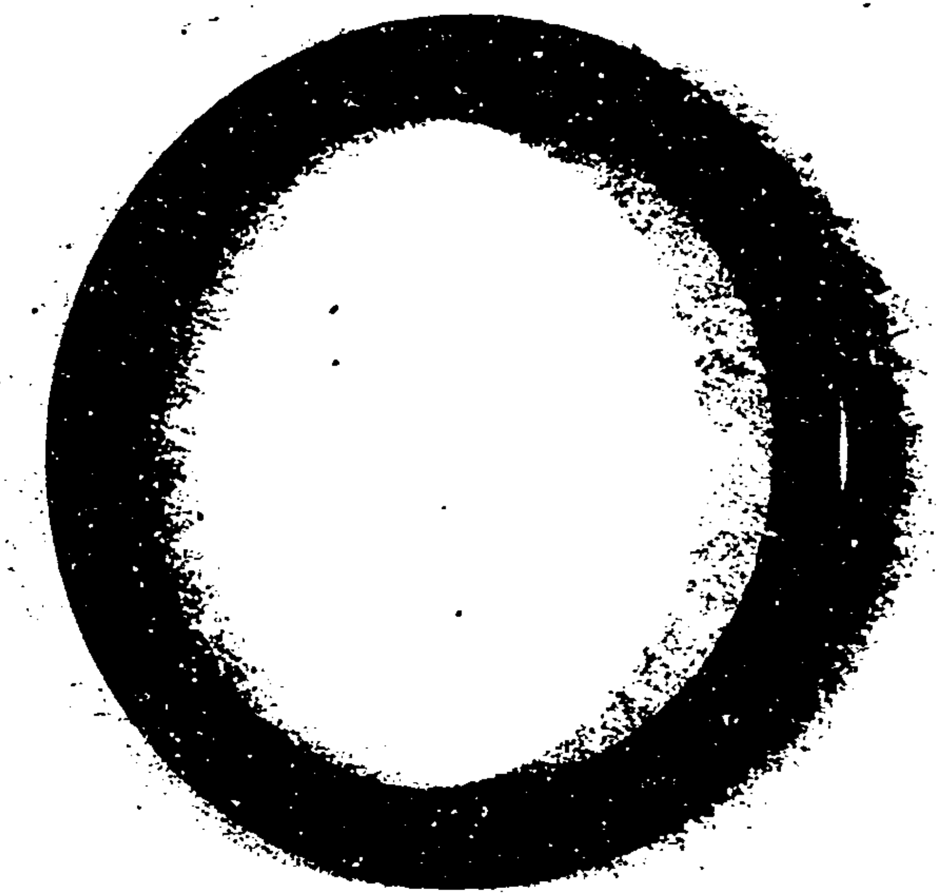

Figure 4-22. Posttest Condition of MS28778-8 O-Ring Installed in PCP Pump (P3)

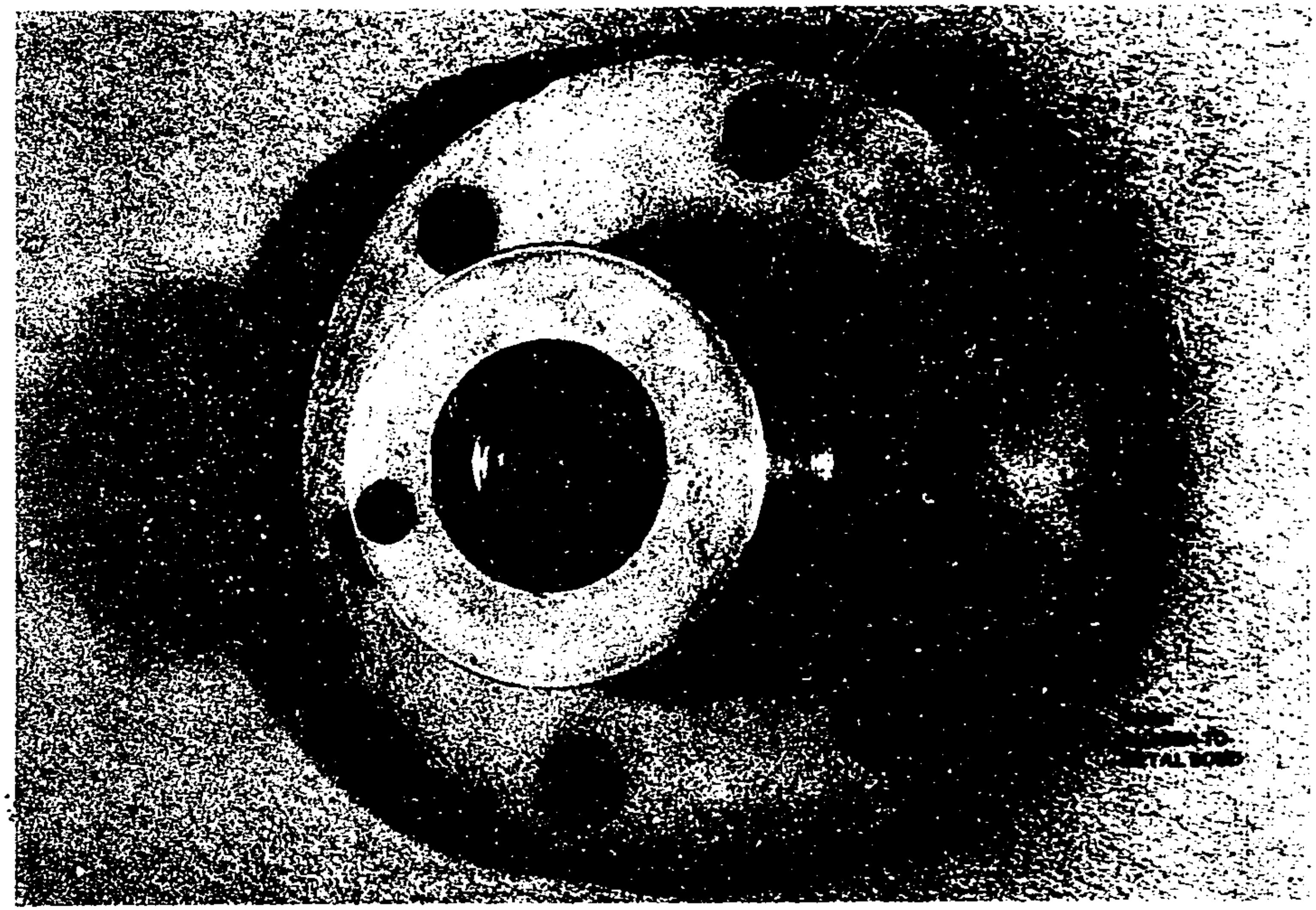

Figure 4-23. Drive Shaft Connection Side of PCP Pump (P3) Diaphragm

parts. There were no tears in the diaphragm face or pieces of it stuck to the pump face. The bond apparently was a vacuum bond.

The pump cover was removed from the drive end after draining the oil. The oil had mixed with the lithium grease and had been pumped back through the cover plate. Part of it was pumped through the rubber plug and part through a hole in the cover plate. This condition had been previously noticed during an inspection of the drive end during the test period on day 33. The grease had been replaced in the harmonic drive section at this time. The hole between the drive section cavity and the oil bath reservoir had been plugged at factory assembly with RTV. The RTV plug was found to be quite loose and had allowed oil to enter and mix with the lithium grease. This mixture was pumped through the cover plate almost to the full capacity of the volume about the plate. Metal particles were found in the bottom of oil bath reservoir as shown in Figure 4-24.

Removal of the motor and the harmonic drive speed reducer mounting flange allowed the remaining oil and lithium grease mixture to be drained from the pump. Two broken screws were found loose in this section. Inspection of the flexspline mounting screws showed two of four screws were still in their mounting holes, although in the process of backing out. These were very loose and upon removal, it was found that all four screws had been broken. Figure 4-25 shows the broken ends still in the shaft. The secured screws broke close to their ends, and the two screws that had broken and fallen out were broken much closer to their heads.

Figure 4-26 shows two cracks in the plastic part of the flexspline. There are eight cracks total but the other six are in the face normal to the metal insert and not as large. There was little, if any, wear on the flexspline or other parts in the drive assembly.

In the design of the pump there appears to be two weak areas. One is the O-ring mounting in the discharge valve assembly. Figure 4-27 shows the O-ring (PCP Part No. 996 or MS 28778-8) as retained by the head and the valve seat. As evaluated by MDAC personnel the 996 O-ring is not sufficiently retained to be protected from backpressure action during intake. Both direct

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Figure 4-24. Metal Particles Wiped from Bottom of PCP Pump (P3) Oil Bath Reservoir

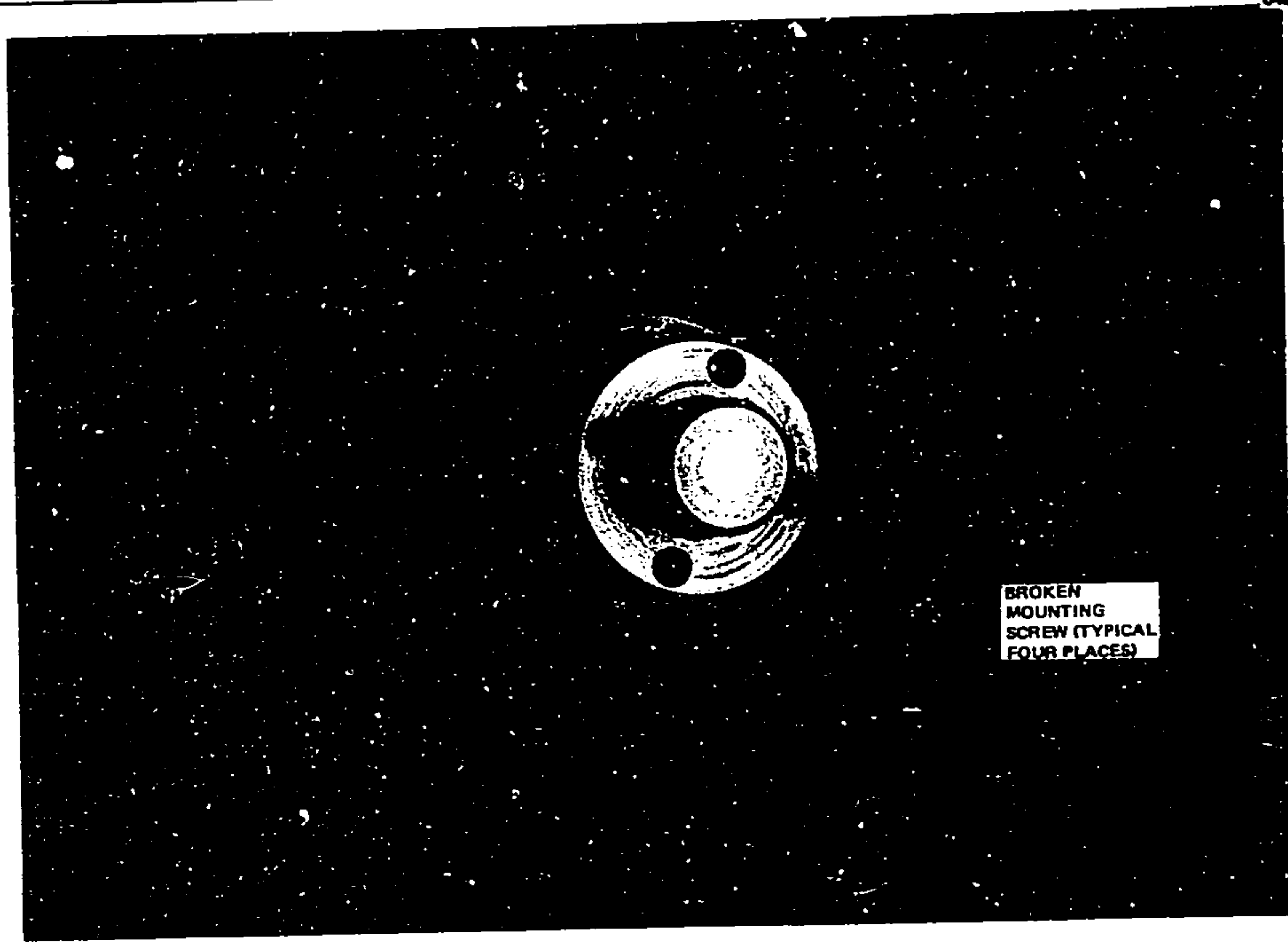

Figure 4-25. Broken Flexspline Mounting Screws in PCP Pump (P3)

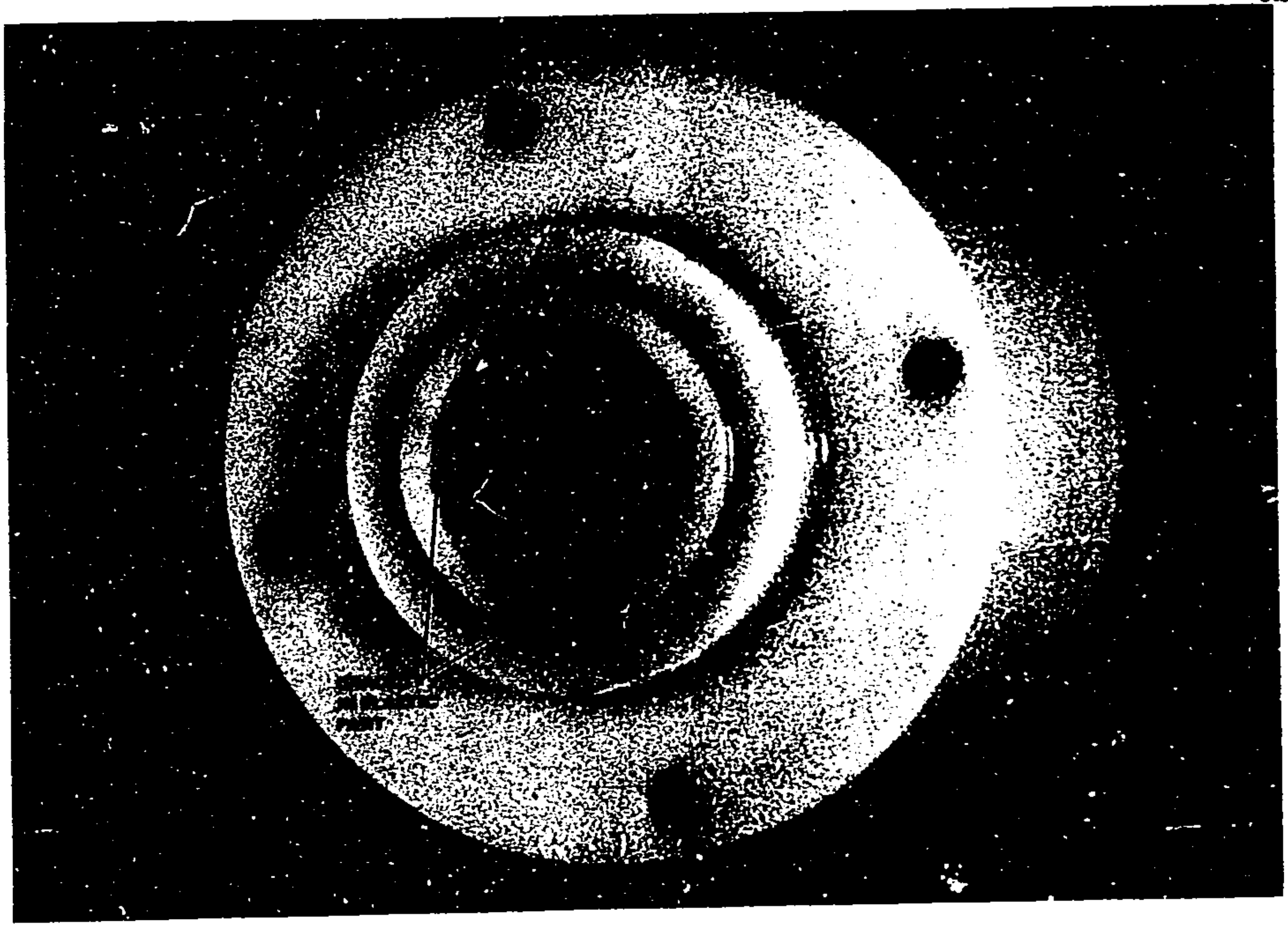

Figure 4-26. Cracked Flexspline of PCP Pump (P3)

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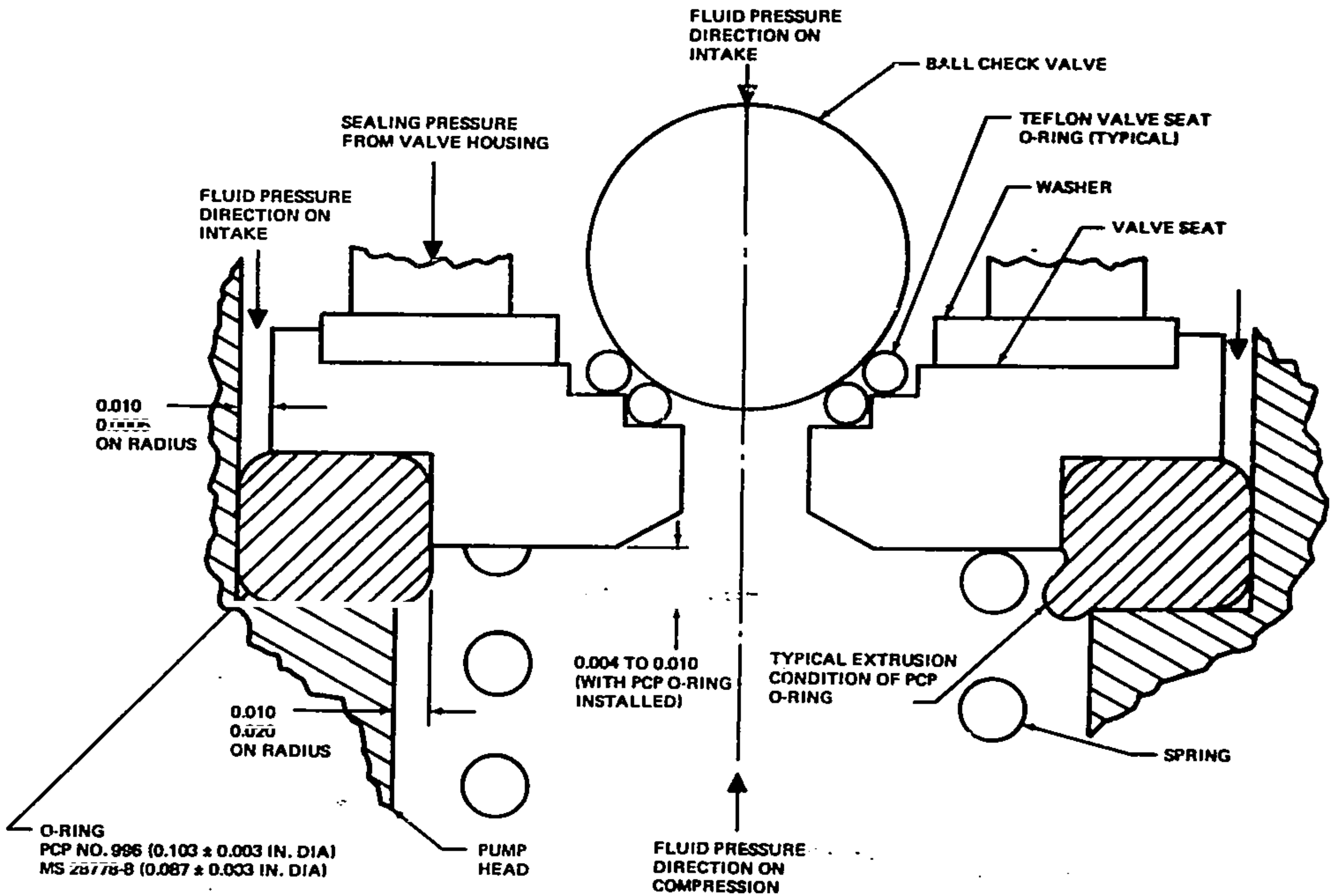

Figure 4-27. Assembly of PCP Pump (P3) Outlet Check Valve

backpressure and the reciprocating of the valve seat could contribute to driving the O-ring off the ledge on which it is seated. This would allow the O-ring to be cut by the bottom corner of the valve seat and, once it is torn sufficiently, to be driven down around the spring.

The cross-sectional area of the O-ring also contributed to its being caught and abraded by the valve seat. Its cross-sectional diameter of 0.103 ± 0.003 in. is too large for the space allowed. The MS-28778-8 O-ring which was installed on day 7 had a cross-sectional diameter of 0.087 ± 0.003 in. and was never dislodged.

Additionally it should be noted that the machined sealing surfaces throughout the head and their parts are finished to approximately 63 RMS at best. General industrial practice and seal and O-ring handbooks recommend a minimum of a 32 RMS finish for O-ring grooves and sealing surfaces.

The second area to be discussed concerns the harmonic drive (see Figure 4-19). The flexspline contained cracks, and its four mounting screws were all broken. The manufacturer's engineering department identified these symptoms as indicative of overheating. The overheating has occurred in a fraction of their pumps since February 1973 due to faulty yokes. The yoke is a part in the drive train, and some of those received in February were defective. The defective yoke prevents the follower bearing from turning and this causes the shaft to overheat. The heat is transferred to the flexspline causing cracking in the plastic and the breaking of the bolts. Several openings and seams in the assembly that are normally sealed also open sufficiently to allow the lithium grease and oil to mix and to be pumped through the cover plate. The clicking noise, which was noticed 8 hours operating time prior to the end of the test, probably occurred with the failing harmonic drive parts. The configuration of the pump drive section allows only minimal parts inspection without considerable disassembly. Therefore the source of the noise could not be detected during the test.

4.4.2.4 Test Performance and Posttest Evaluation of the Haskel Pump (P4)
The Haskel pump, Model MS-12, Serial No. 1272 122, is a reciprocating, packed plunger, direct air actuated pump. A cross-sectional view of this

pump is shown in Figure 4-28. A summary of the pump operation during the test is shown in Table 4-8. This pump operated for 321.55 hours during the test prior to final failure. This pump was shut down for 67.667 hours due to pump malfunctions and for 14.817 hours due to facility malfunctions.

The daily average flow rate of this pump is shown in Figure 4-29. This pump never achieved the desired 0.056-gpm flow rate. The average flow during the test was 0.0456 gpm and the maximum average daily flow was 0.0558 gpm, which was achieved on day 12 after the pump had been rebuilt.

This pump should have operated at 800 psig with an air pressure of 67 psig (the pump had an effective air side to liquid side area ratio of 12:1). However, the design flow rate could not be obtained and the air pressure was increased gradually to 95 psig in an attempt to achieve the design flow rate. Operation at this higher pressure required modifications to the test stand to incorporate a high-pressure control switch to limit the normal operating pressure to 840 psig maximum.

The Haskel pump had two characteristics that diminished its effectiveness. The plunger packing would not function in a static condition and water would rapidly leak into the drive section and out the lower exhaust port unless the supply valve was closed as the pump stopped operating. This occurred soon after the pump was installed, and did not seem to be a function of wear. The second problem was the occasional sticking of the pilot valve which would stop the pump. The pump would begin reciprocating again when the air cap (Figure 4-28) was struck sharply. This occurred three times during the test. The third time it happened (on day 9) the discharge pressure was also fluctuating from 250 to 400 psig. The pump was removed at this time and disassembled.

The liquid outlet check valve assembly was inspected and the edges of the guide were smoothed and the valve assembly was reassembled. The main body of the pump was disassembled and soap residues were found in and above the packing, but the liquid inlet check valve parts and the air barrel were quite clean. There was some scoring of the plunger. The air drive end